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A DESCRIPTION

OF THE

DIFFERENTIAL EXPANSIVE PUMPING ENGINE,

WITH USEFUL NOTES,

BY

HENRY DAVEY, C.E., M.I.M.E., F.G.S.

HATHORN, DAVIS, CAMPBELL, & DAVEY,

ENGINEERS, LEEDS.

MAKERS OF—

- THE DIFFERENTIAL PUMPING ENGINE,
- HYDRAULIC PUMPING ENGINES,
- CORNISH ENGINES,
- DIFFERENTIAL BLOWING ENGINES,
- COMPOUND ROTATIVE ENGINES,
- THE SEPARATE CONDENSER,
- HYDRAULIC MACHINERY,
- MINING PLANT OF ALL KINDS, AND
- MACHINERY FOR WATER SUPPLY, IRRIGATION, &c.

* * *

Leeds:
PRINTED BY McCORQUODALE AND CO., BANK STREET.

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1875.

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1903

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The above illustrations have been prepared from the drawings of engines which have been made, and some of them are accompanied with tables of various sizes for which patterns exist. References can be given to large numbers of engines at work.

DESCRIPTION

OF THE

"DIFFERENTIAL EXPANSIVE"

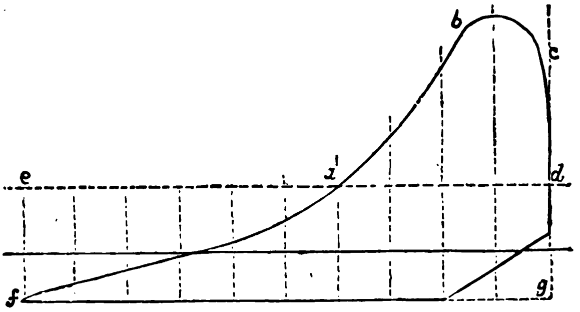
PUMPING ENGINE.

◆

THE Differential engine exists in two distinct types,—viz., the single cylinder and the compound engine; the latter admitting of being worked with very high pressure steam and high degrees of expansion, is capable of realising the greatest economy of fuel. The chief peculiarity in the invention, is the simple manner in which the engine is made perfectly safe in working under all conditions of load, automatically and instantly varying its supply of steam with very minute increase or decrease of resistance; the distribution of steam being such, that the pumping is performed without shock even when the pressures suddenly and greatly vary.

In designing pumping machinery, as also in designing steam machinery of all kinds, the three great questions which should be thoroughly and relatively considered are—economy of fuel, economy of maintenance, and economy of construction. The importance of economy in fuel was never more keenly realised than at the present time, notwithstanding the great advances which have been made to reduce the consumption. Our steam engines consume annually 37,000,000 tons of coal, which at the present moment may, perhaps, be reckoned at 15s. per ton, representing over £27,000,000 sterling; an economy of 25 per cent. would therefore effect a saving of nearly £7,000,000 annually. With colliery pumping engines it is not unusual to find a consumption of from 12 lbs. to 14 lbs. of coal per horse-power per hour. A good compound engine will work with less than one-fourth that amount of fuel. The saving to be effected on 400 horse-power of actual work by the substitution of a compound engine for a non-expansive engine would be at the lowest estimate 36 tons in twenty-four hours, which, taken at 5s. per ton at the pit's mouth, would amount to the modest sum of £2,700 per annum.

It is well known that the leading principle of economy in the steam engine is that of expansion of steam, and the introduction of the second cylinder to form a compound engine gives no theoretical advantage, but is purely a practical question. By the introduction of the second cylinder the maximum strain thrown on the piston and its attachments is greatly reduced, and a greater uniformity of motion and strain is secured. In non-rotative engines, expansion cannot be carried to a great extent in a single-cylinder, as will be seen from the following analysis:—The diagram, which is shown below, is taken from a Cornish engine. The line drawn horizontally through the diagram cuts the expansion line at the point of average pressure, making the area *a, b, c, d,* equal to the area *a, e, f.*



It is evident that if the diagram was produced by a constant force overcoming a constant resistance it would take the form of the parallelogram *e, f, g, d,* but under such circumstances the steam would not be expanded. In the Cornish engine the load lifted is a constant quantity, and that steam can be at all expanded on the piston is due to the fact that the moving load is a mass of matter and not merely an opposing force. The expansive force of the steam can be utilized because the mass in being changed from a state of rest to that of quick motion, absorbs or stores up an amount of energy which is expended during the second change from quick motion to the former state of rest. In other words, during the first change, energy, represented by the portion of the diagram *a, b, c, d,* is accumulated, and this energy, which is also represented by the area *a, e, f,* is during the second change expended. The energy of the mass equals $\frac{Wv^2}{2g}$, and the energy represented by the area *a, b, c, d,* equals the mean pressure of that portion of the diagram multiplied by the distance *a, d,* and the resultant multiplied by the area of the piston, making in the present example 150 foot-tons. The load on the piston is not more than 50 tons, and if the engine had no mass to set in motion beyond the column of water to be lifted, the required velocity of that mass necessary to give it the required energy would be beyond all practical limit. On the other hand, if the velocity of the mass is limited to 4 feet per second, then the required mass would be about 600 tons, although the expansion is only about 4·5 to 1.

If this engine were to be made double-acting and employed to lift water on the steam stroke it would be expedient, for practical as well as theoretical reasons, to limit the piston speed to 4 feet per second, making it necessary, as the author has already shown, to provide a moving mass of 600 tons, but with the compound engine, working under similar conditions, only a very small moving mass would be necessary, because the variation in the effective pressures during the stroke would be comparatively very little. An expansion of eight times in a single cylinder would give a variation of 8 to 1, but in a compound differential engine the force is nearly constant. The Cornish engine is a very costly machine; it is only single acting, and it requires a great moving mass to make it practicable to employ a high degree of expansion; nor can it be made double acting without either making it necessary to increase the mass beyond a practical limit or to employ a low ratio of expansion involving a great waste of fuel.

To meet these difficulties the author designed the compound differential expansive engine, in which expansion may be carried to a great extent with a piston speed not too great for actuating pumps during both strokes, nor requiring a heavy moving mass, making it practicable to secure economy of fuel with a comparatively small outlay of capital, and greatly reduced expenditure in maintenance.

For a graphic illustration of the Differential gear see its application to Cornish engines, pages 38 and 35. In the Differential engine, the main slide or other valves receive their motions from a lever to the centre of which they are connected. This lever receives two motions, one derived from the main moving parts of the engine by another lever of the first order, which, receiving the full motion of the piston at the long end, imparts from its short limb the amount of motion suitable to the working of the valve; and another motion derived from a subsidiary piston. This subsidiary piston receives its motion from the steam in the main slide chest by means of a small valve, and gives motion to a cataract piston working in a cylinder filled with water which escapes from side to side through small openings that can be regulated at pleasure. This cataract regulates the speed of the subsidiary piston, and consequently the motion of the end of the lever to which it is attached.

The action of this gear upon the motion of the engine will be best understood by an illustration: Suppose the main piston to be at rest at one end of the cylinder, then to start the engine, steam would be admitted into the subsidiary cylinder, and motion would be communicated to the valves, and the engine would commence its stroke; as it moves, however, it is giving motion to the valve lever in a contrary direction to the motion communicated by the subsidiary piston and cuts off the steam. The main valve, therefore, has a differential motion, compounded of the motion derived from the direct action of the main piston, and an opposite motion from the subsidiary piston. Now, the motion of this subsidiary piston is rendered constant by means of the cataract. Seeing, then, that the cataract end of the lever has a constant motion independent of the engine itself, and that the other end must needs have a varying motion depending on the varying load on the engine, then the resultant motion of the main valve, being taken from the centre of the lever and compounded of a varying and a constant motion, must also vary with every variation in the motion of the main piston.

The main slide valve spindle is attached to the centre of a differential lever, one end of which derives its motion from the engine, and reciprocates with the steam piston, whilst the other end is actuated by means of a subsidiary piston, caused to reciprocate in equal times, by means of a double-acting cataract governor. The movement of the cataract end of the lever is constant, and independent of the motion of the main piston of the engine, and has necessarily a lead in advance of that of the opposite end, at the commencement of each stroke. With a constant load on the engine the lead will be constant, and the cut-off constant; but the slightest variation in the load causes a corresponding instant variation in the lead, and as a consequence the cut-off takes place earlier or later, as the load is decreased or increased. The engine thus automatically varies the expansion to suit the varying conditions of resistance.

The subsidiary piston must necessarily move the slide valve through a distance equal to the lap of the main valve or that of the expansion valve, before the engine can commence its stroke, and thus a pause is produced as in the Cornish engine, and a lead is given to the exhaust side causing the spent steam to be exhausted, or partly so, before the engine commences its next stroke, and thereby preventing back pressure.

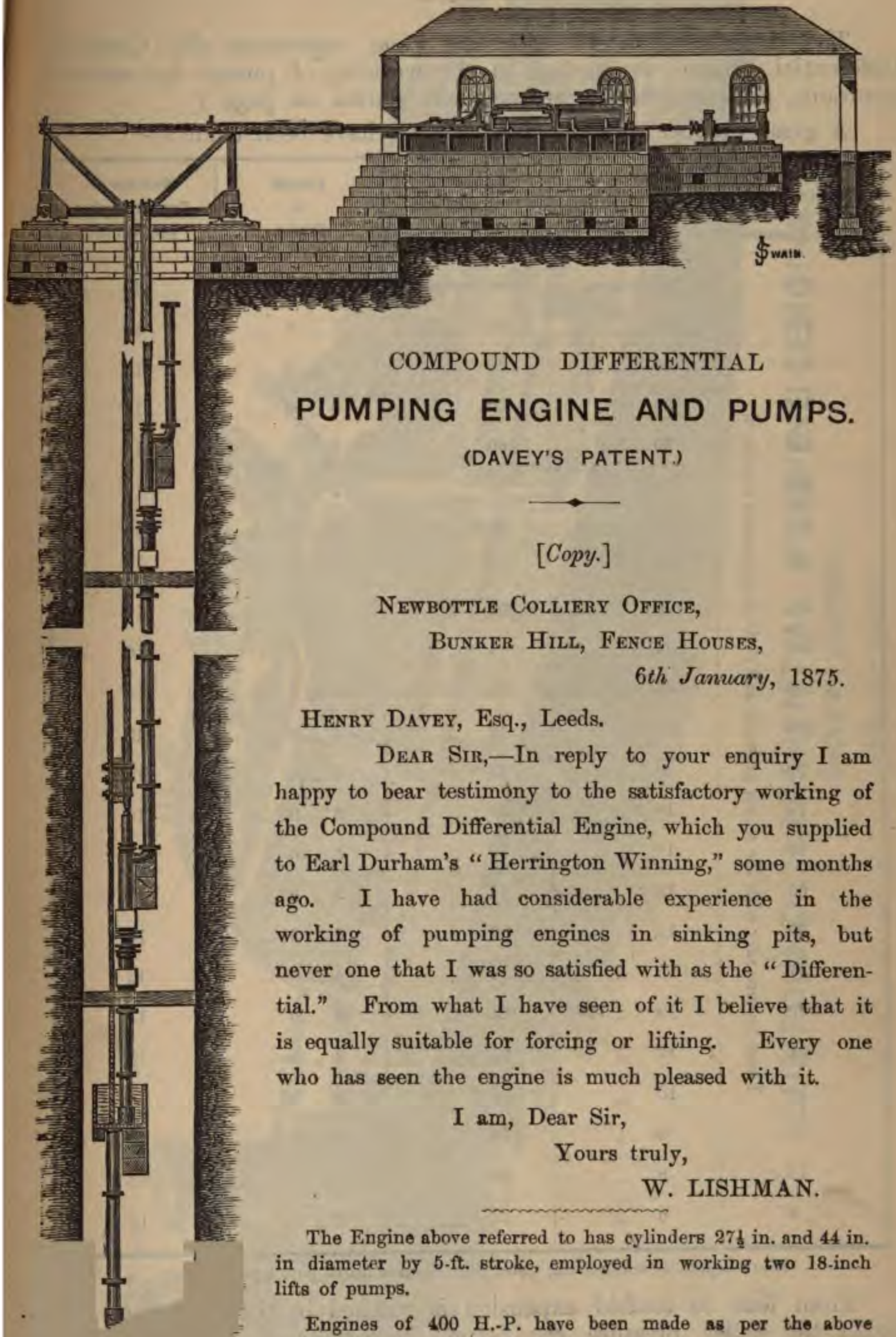
With ordinary pumping engines there is a certain amount of shock produced consequent on the closing of the pump valves, which is more or less according to design and proportions of the engine and pumps, the heights of the lift, &c. The shock acts prejudicially, in injuring the valves, shortening the durability of the engine, and rendering it liable to accident, such as bursting the pump, pipes, &c. To prevent such shocks is of great consequence, especially in high lifts. The motion of the piston of the "Differential Expansive" Engine is such that a pause is produced at the completion of each stroke, during which time the valves are allowed to fall by their own weight alone to their seats, preventing the possibility of a shock from sudden closing under pressure, and also preventing a loss of efficiency from slip.

The Compound Differential Engine is superior to the Cornish Engine, inasmuch as a greater expansion of steam can be realized, with the consequent economy of fuel; also because it is safe in working against variable loads; requires lighter pit work and less costly buildings and foundations. When applied directly to forcing pumps its advantages are equally obvious, for economy of fuel is secured, combined with the gain effected in dispensing with spear rods and heavy pit-work.

It is not only desirable to have an economical engine, but it is also *most important* to apply the engine to an efficient pump. Piston pumps are to be condemned (save in exceptional cases), especially in mines. A perfect pump for a permanent engine should have no internal packing, or wearing surface, which may be affected by grit or sand in the water, and a pump to fulfil these conditions must have a plunger or plungers provided with external stuffing-boxes, such as those illustrated on pages 16, 17, 20, and 32. A piston pump may be allowing a large quantity of water to pass the piston every stroke without its being detected; but with a plunger pump the slightest leakage is at once seen. The stuffing-boxes of pumps should be packed with good hemp gasget steeped in melted tallow. There is no better packing.

Pumping in Sinking Pits.

The most trying condition under which a pumping engine can be worked is that of pumping from pits during sinking operations. That the "Differential" has proved itself suitable in such a case will be seen from Mr. Lishman's report, printed on the next page. Several engines have been applied to sinking pits as permanent engines, and their advantages over the Cornish and rotative pumping engines are obvious.



COMPOUND DIFFERENTIAL

PUMPING ENGINE AND PUMPS.

(DAVEY'S PATENT.)

[*Copy.*]

Newbottle Colliery Office,
Bunker Hill, Fence Houses,
6*th January*, 1875.

Henry Davey, Esq., Leeds.

Dear Sir,—In reply to your enquiry I am happy to bear testimony to the satisfactory working of the Compound Differential Engine, which you supplied to Earl Durham's "Herrington Winning," some months ago. I have had considerable experience in the working of pumping engines in sinking pits, but never one that I was so satisfied with as the "Differential." From what I have seen of it I believe that it is equally suitable for forcing or lifting. Every one who has seen the engine is much pleased with it.

I am, Dear Sir,
Yours truly,
W. LISHMAN.

The Engine above referred to has cylinders 27½ in. and 44 in. in diameter by 5-ft. stroke, employed in working two 18-inch lifts of pumps.

Engines of 400 H.-P. have been made as per the above illustration.

THE COMPOUND DIFFERENTIAL PUMPING ENGINE.

The engraving on the opposite page represents the Compound Differential Engine, as applied to the working of pumps by means of quadrants, according to the arrangement shown on page 7.

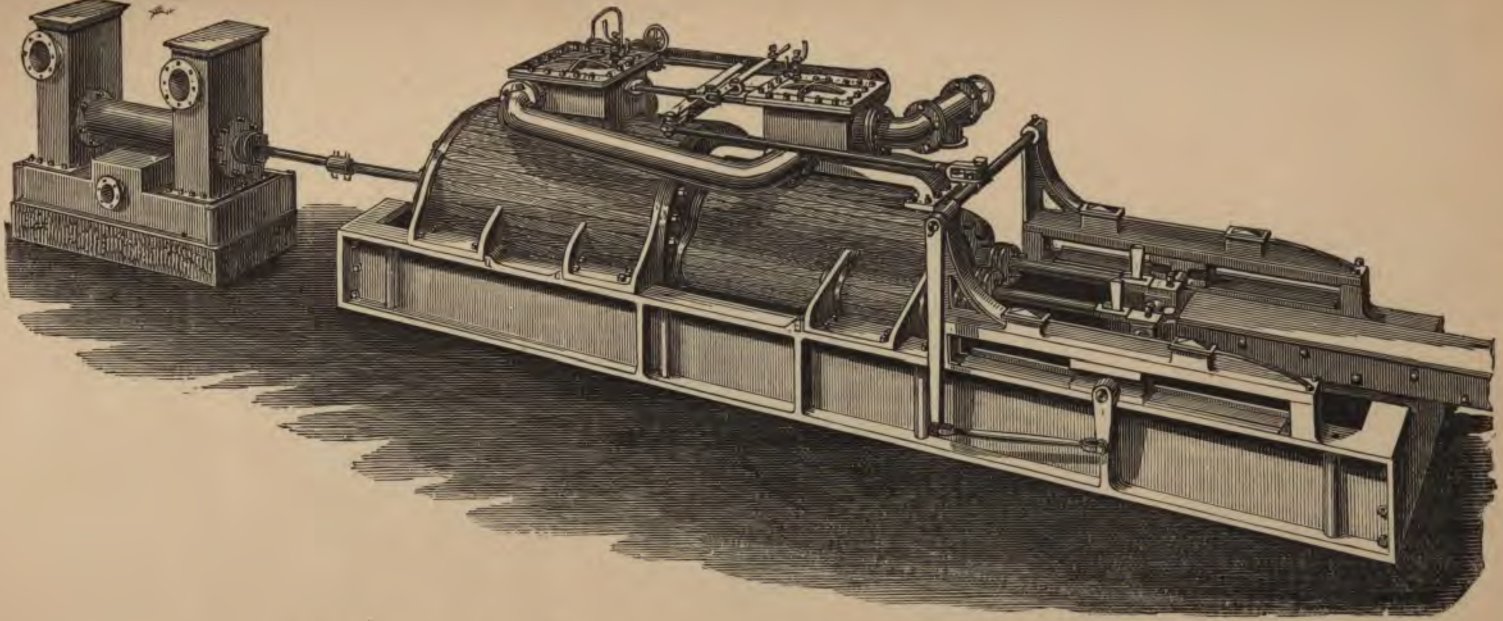
A great number of the following sizes have been made:—

Approximate Effective Horse-power for 50 lbs Boiler Pressure.	Diameter of High Pressure Cylinder.	Diameter of Low Pressure Cylinder.	Length of Stroke.	Number of Double Strokes per Minute.
310	34 inches.	64 inches.	8 ft. 0 ins.	10
293	34 ,,	64 ,,	7 ,, 6 ,,	10
260	35 ,,	60 ,,	7 ,, 0 ,,	11
246	35 ,,	60 ,,	6 ,, 0 ,,	12
246	30 ,,	60 ,,	6 ,, 0 ,,	12
230	30 ,,	56 ,,	7 ,, 0 ,,	11
185	30 ,,	52 ,,	6 ,, 0 ,,	12
142	30 ,,	50 ,,	5 ,, 0 ,,	12
142	25 ,,	50 ,,	5 ,, 0 ,,	12
154	24 ,,	44 ,,	7 ,, 0 ,,	12
135	24 ,,	44 ,,	6 ,, 0 ,,	12
110	24 ,,	44 ,,	5 ,, 0 ,,	12
90	20 ,,	40 ,,	5 ,, 0 ,,	12
74	18 ,,	36 ,,	5 ,, 0 ,,	12
70	16 ,,	32 ,,	6 ,, 0 ,,	12
63	16 ,,	32 ,,	5 ,, 0 ,,	13
55	15 ,,	30 ,,	5 ,, 0 ,,	13
47	15 ,,	30 ,,	4 ,, 0 ,,	14
30	12 ,,	24 ,,	4 ,, 0 ,,	14
24	12 ,,	24 ,,	3 ,, 0 ,,	15
17	10 ,,	20 ,,	3 ,, 0 ,,	15
10	8 ,,	16 ,,	3 ,, 0 ,,	15
10	8 ,,	16 ,,	2 ,, 6 ,,	18
6	6 ,,	12 ,,	2 ,, 6 ,,	18

These powers vary with the steam pressure and ratio of expansion employed.

Allow 2 gals. of condensing water per effective horse-power per minute.

From four to tenfold expansion is realised in these Engines. The higher the boiler pressure the better the results obtained as regards economy of fuel.



COMPOUND DIFFERENTIAL PUMPING ENGINE.

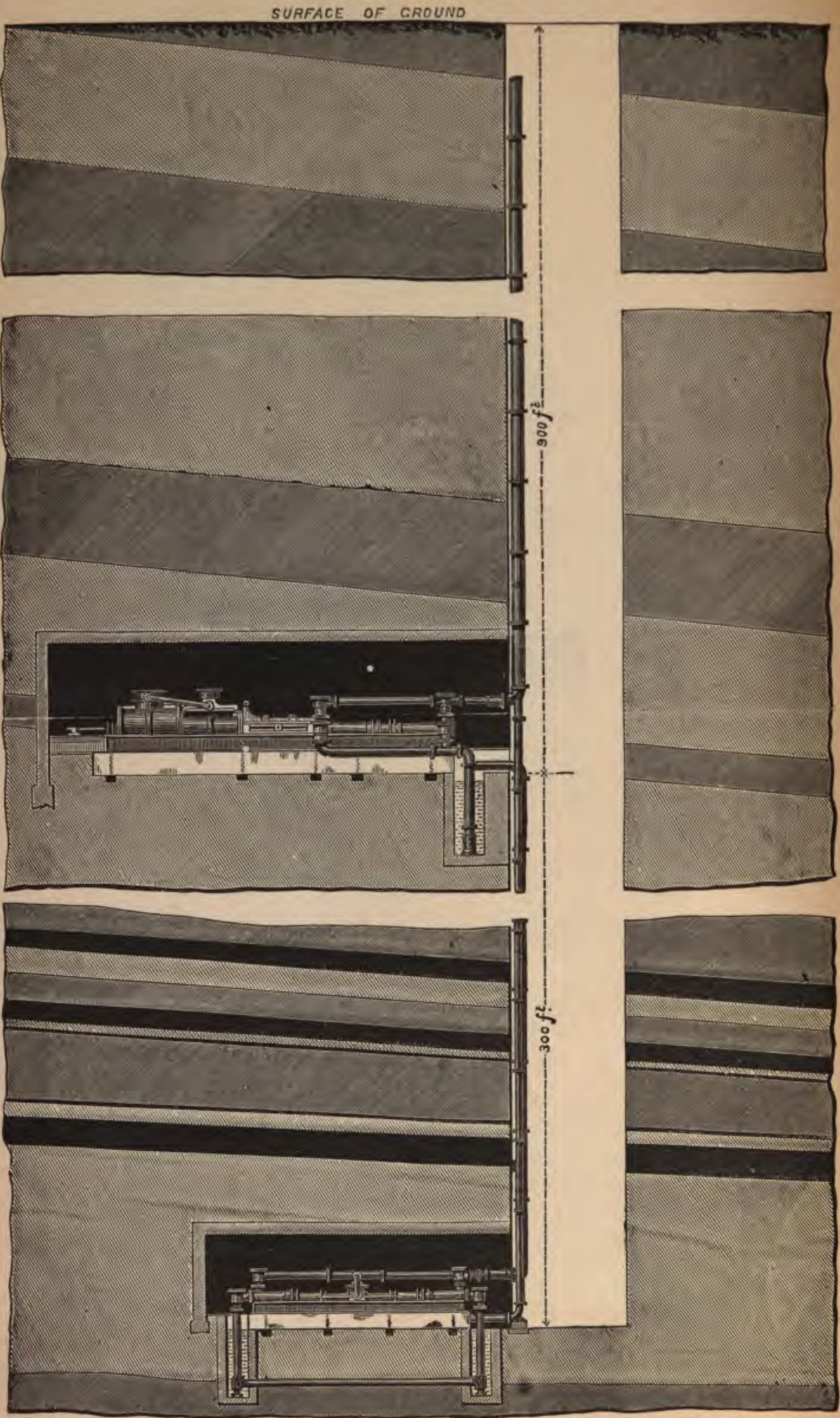
(DAVEY'S PATENT.)

(See also page 7.)

THI

THE NEW Y

DAVEY'S SYSTEM OF UNDERGROUND PUMPING ENGINES.



DAVEY'S SYSTEM

OF

UNDERGROUND PUMPING ENGINES

FOR

COLLIERIES AND OTHER MINES.

The system illustrated on the opposite page is being carried out on a large scale in a colliery 1,200 feet deep. At a point in the pit 900 feet from the surface are placed the pair of Compound Differential Engines, pumps, and separate condenser, further illustrated on pages 16 and 17. The engines have cylinders 35 and 60 inches diameter, by 6 feet stroke. The pump rams are of gun metal, 12½ inches diameter. At the bottom of the pit, 300 feet below the main engines, are placed a pair of hydraulic pumping engines, further illustrated on page 23. These engines are to lift 1,000 gallons per minute to the main engines. The main engines force the water to the surface and supply power through the column to the hydraulic engines. By this system the main engines are kept out of danger of flooding. The hydraulic engines will work under water, and can be actuated from the main engine room. As further security, they could be placed in a water-tight chamber, accessible from the main engine room through a water-tight staple. By such means the hydraulic engines could be under repair even when the water rose to the main engines, 300 feet up the shaft.

The economy of this system of draining mines is very great. Although the steam engines when placed underground consume a little more steam than if placed on the surface,* the useful effect is largely increased by dispensing with spear rods, &c.; therefore, the effective work done by a given weight of steam is increased, whilst the cost of maintenance is immensely reduced.

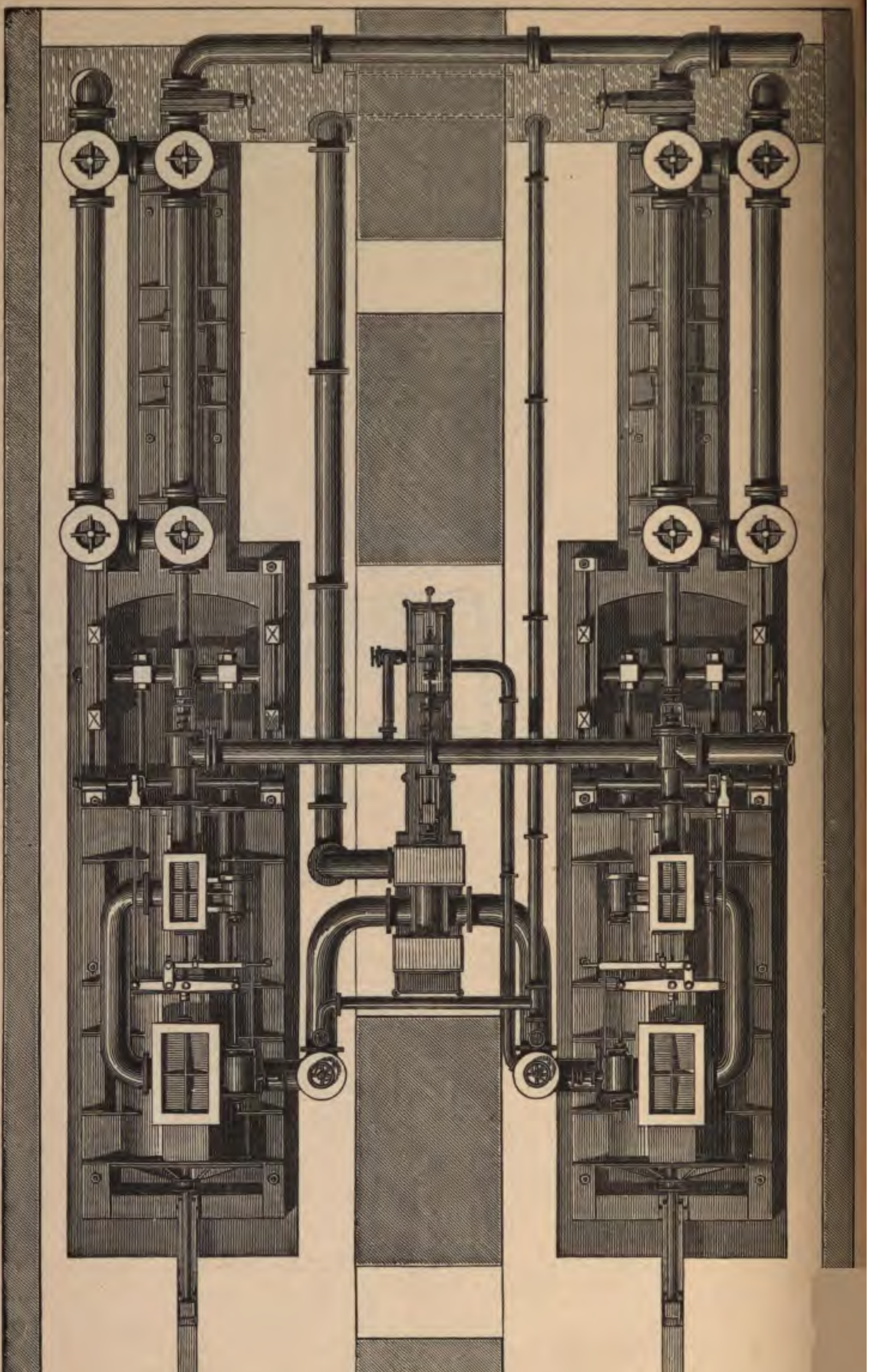
DRAINING DIP WORKINGS.

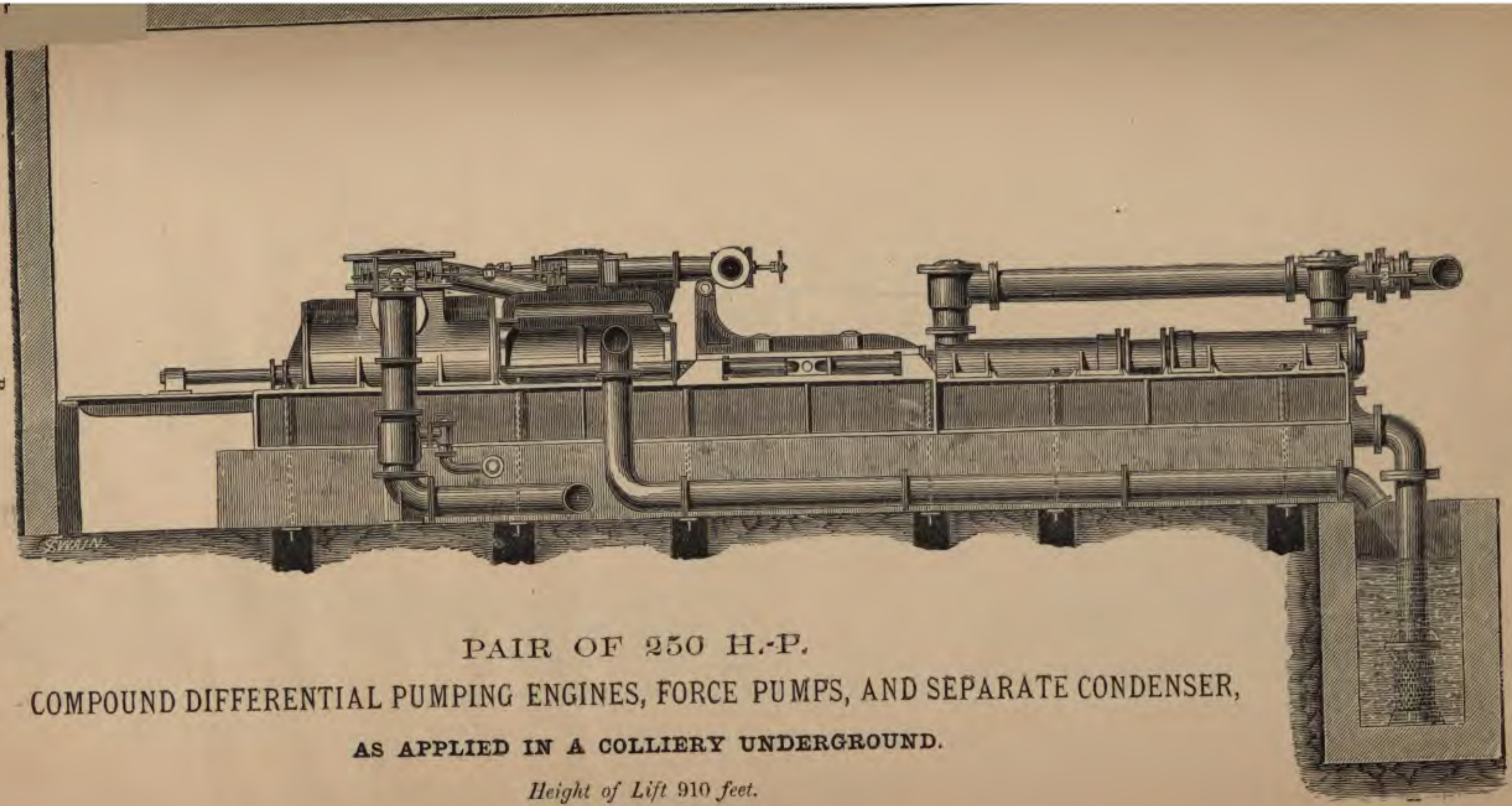
Dip workings are drained by means of hydraulic engines applied on the same principle as those illustrated on the opposite page, and a great number are working very successfully in various collieries. The useful effect is about 80 per cent. of the power applied.

---

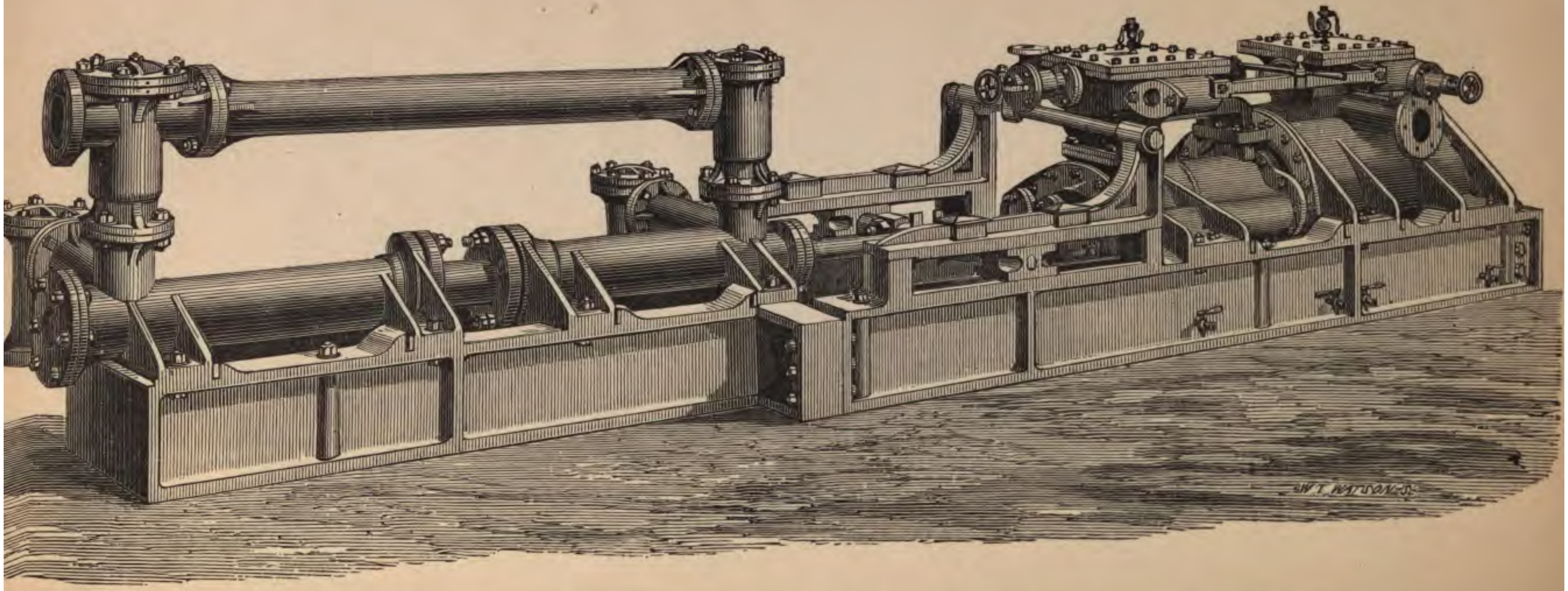
* The loss from condensation in a well clothed pipe may be taken approximately as 1 lb. of steam per indicated H.-P., per hour, for each 100 ft. of steam pipe.

Note.—A stuffing-box expansion joint should be provided in the steam pipe allowing an expansion of 2¼ in. for each 100 ft. in length.





PAIR OF 250 H.-P.
COMPOUND DIFFERENTIAL PUMPING ENGINES, FORCE PUMPS, AND SEPARATE CONDENSER,
AS APPLIED IN A COLLIERY UNDERGROUND.
Height of Lift 910 *feet.*



W.T. WATSON SC.

COMPOUND DIFFERENTIAL PUMPING ENGINE AND FORCE PUMPS.

(DAVEY'S PATENT.)

COMPOUND DIFFERENTIAL PUMPING ENGINES AND FORCE PUMPS.

(See also pages 16, 17 *and* 48.*)*

Approximate Effective Horse-power.	Diameter of High-pressure Cylinder.	Diameter of Low-pressure Cylinder.	Length of Stroke		Number of Strokes per Minute.	Diameter of Pump Ram.	Number of Gallons raised per Minute.	Height of Lift.	
	Inches.	Inches.	Ft.	Ins.		Inches.		Feet.	
293	34	64	7	6	10	...	...	...	Allow 2 gallons of condensing water per effective horse-power per minute.
260	35	60	7	0	11	...	...	...	
*246	35	60	6	0	12	*12½	700	910	
246	30	60	6	0	12	...	...	...	
230	30	56	7	0	11	...	...	...	
185	30	52	6	0	12	...	...	...	
142	30	50	5	0	12	†12	500	800	
142	25	50	5	0	12	...	...	...	
154	24	44	7	0	11	...	...	...	
135	24	44	6	0	12	11	560	700	
110	24	44	5	0	12	11½	500	600	
90	20	40	5	0	12	9	300	700	
74	18	36	5	0	12	17	1000	180	
70	16	32	6	0	12	...	...	...	
63	16	32	5	0	12	17	1000	150	
55	15	30	5	0	12	9	300	400	
47	15	30	4	0	14	12½	520	240	
30	12	24	4	0	14	...	...	...	
24	12	24	3	0	15	6	100	400	
17	10	20	3	0	15	...	...	...	
10	8	16	3	0	15	...	...	...	
10	8	16	2	6	18	...	...	...	
6	6	12	2	6	18	...	...	...	

* For illustrations of a pair of these engines see pages 16 and 17.

† This engine is provided with a surface condenser.

Note.—These engines are provided with surface or injection condensers, and are so designed that Pumps of any size can be fixed to them, provided the strokes are the same. The Pumps are of the double-acting ram-variety, and are not affected by sand or grit, which rapidly destroys piston pumps.

THE DIFFERENTIAL HYDRAULIC PUMPING ENGINE.

(DAVEY'S PATENT.)

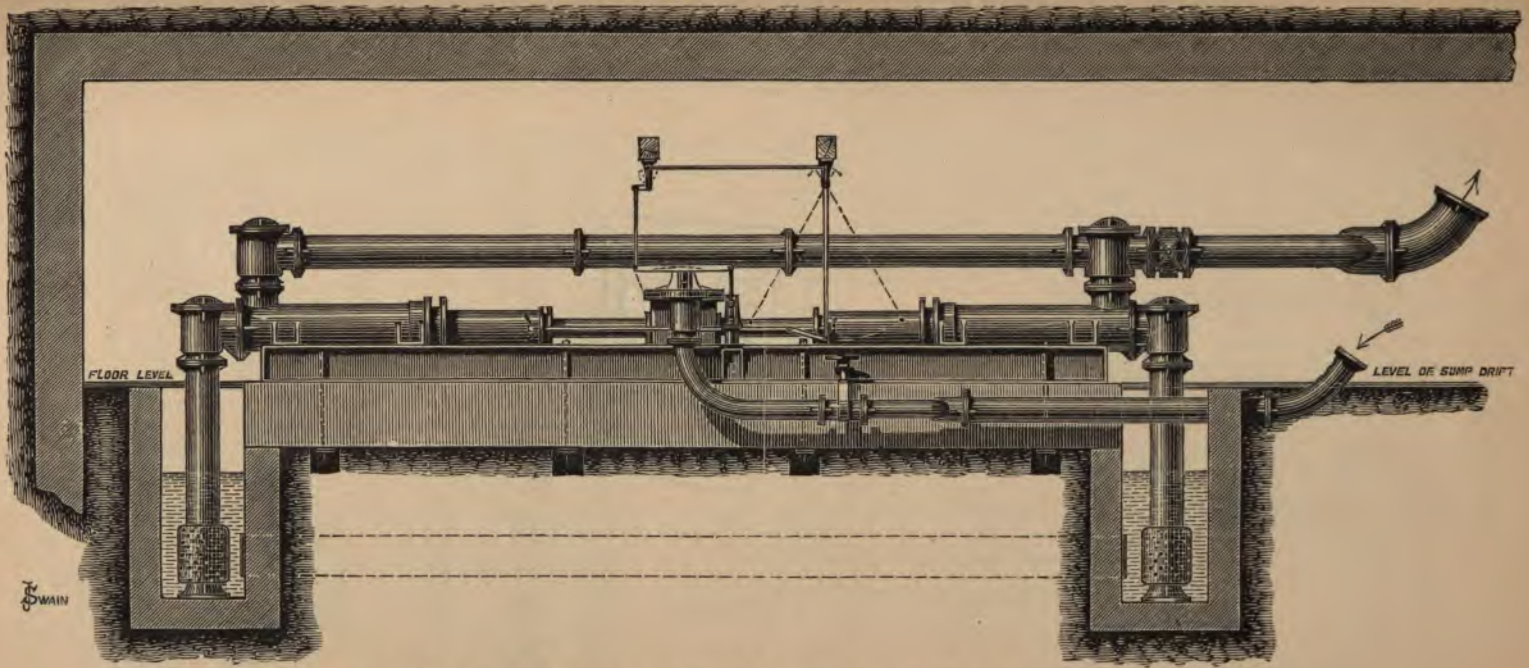
This engine has a variety of applications. It is employed in pumping from the bottom of a mine to a point some distance up the pit where the main pumping engines are situated, as illustrated on page 12; or it may be worked from a natural head of water, or from an accumulator. The principle of its action is that of employing water at a given head to raise a larger quantity against a less head. In hilly mining districts, water may be accumulated at a high elevation, and conducted to the mine in pipes, there employed in one of those engines to pump the water from the mine. In this way the cost of steam power may often be saved. These engines are also suitable for accumulating power and transmitting it from one point to another far distant. For example—water flowing from an elevation on one side of a valley may be made to pump water to a higher elevation either on the same or on the opposite side, and there stored for use or employed in giving motion to machinery, &c., &c.

The illustration represents one of a pair of engines made to raise 1,000 gallons per minute 300 feet high. In this engine there is no piston, neither is there any internal packing or slide valve to get out of order. The packing is all applied in stuffing-boxes. The valves are gun-metal mitre valves with large areas, producing no throttling, and will work under the heaviest pressures without excessive wear. The engine is under self-control by means of the differential gear.

Draining Pits or Shafts during Sinking Operations.

(DAVEY'S PATENT SYSTEM.)

To be able to drain pits without the delays incidental to the use of spear-rods is an advantage which will be fully appreciated by mining engineers. The author's system is that of employing hydraulic power, in a very simple manner, to the purpose. Pits may be drained on this system with the minimum amount of stoppages, and the plant used may be retained as permanent plant after the sinking has been completed.



THE DIFFERENTIAL HYDRAULIC PUMPING ENGINE.

(DAVEY'S PATENT.)

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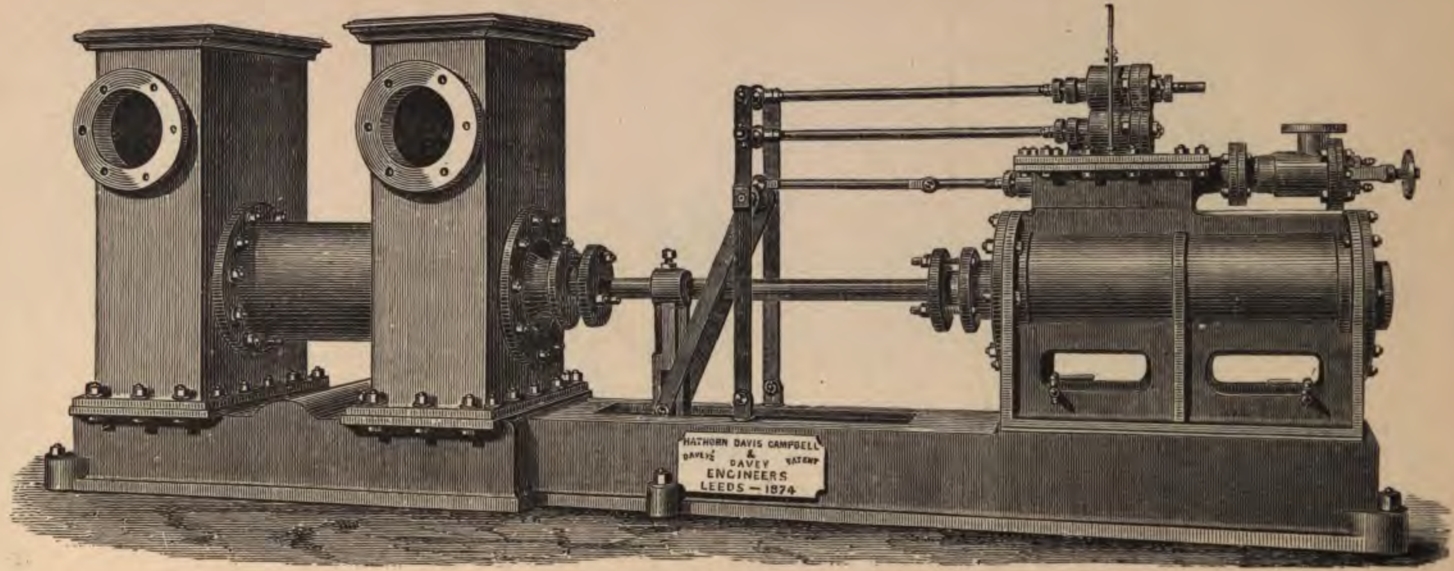

THE SEPARATE CONDENSER.

(DAVEY'S PATENT.)

THE SEPARATE CONDENSER.

This Condenser has been designed to meet the want felt at the present time for a Condenser capable of being easily applied to existing non-condensing engines, to economise fuel and to increase the power. By its application from 30 to 40 per cent. increased power is available, and from 30 to 40 per cent. of fuel is saved.

The Air Pump of the Separate Condenser is worked by means of a small differential steam engine. In applying the Condenser to an existing engine all that is necessary is to fix it in any convenient place and lead the exhaust and a small steam pipe to it.

For Winding Engines the Separate Condenser is invaluable, because the vacuum is not at all impaired by the repeated stopping of the engine. When this Condenser is applied to engines situated underground in mines, a hydraulic engine is sometimes used to work the air pump. The power being obtained from the column or the tubbing.

The advantages claimed for this Condenser are—

1st.—*Economy of Fuel and Increase of Power.* The additional average pressure available on the piston of the engine by the use of the Separate Condenser may be taken as 12 lbs. per square inch, consequently, if the engine is not required to do additional work the average steam pressure may be reduced 12 lbs. This, in cases where the average steam pressure is not high, would represent a saving of 40 per cent. of steam, and in the majority of cases the saving would much exceed 30 per cent.

2nd.—The vacuum is maintained unimpaired when the engine is irregular in its working and when it ceases to work, which makes it peculiarly advantageous for winding engines.

3rd.—One Condenser may be made to condense the steam for several independent engines.

4th.—The Condenser being entirely distinct from the engine, it is more accessible, is easier managed, and in the case of requiring repairs it does not prevent the engine being worked as a non-condensing engine.

5th.—A Boiler-Feed Pump may be combined with the Separate Condenser, arranged to feed the boilers from the hot-well.

In writing for estimates the following particulars of the engine or engines to which the Condenser is to be applied should be stated, viz.:—

Number of Cylinders .. ..
Diameter of Cylinders.. .. ..
Length of Stroke .. ..
Greatest number of Strokes per minute .. ..
Average pressure of Steam during the Stroke ..
Type of Condenser, *Surface or Injection* .. ..
Is the Condenser required to reduce the consumption of coal, or to increase the power of the engine?

THE DIFFERENTIAL STEAM PUMP.

The engraving on the opposite page represents the Differential Steam Pump, fitted with a condensing apparatus,* by means of which the exhaust steam is condensed and economy of fuel effected. In writing for prices it should be stated whether the condenser is required or not. The following table gives particulars of these pumps of various sizes :—

Diameter of Steam Cylinder.	Diameter of Pump.	Length of Stroke.	Number of Strokes per minute.	Number of Gallons per hour.	Horse-Power of Boiler supplied, allowing 2 cubic ft. per H.-P. per hour.	Diameter of Steam Pipe.	Diameter of Exhaust Pipe.	Diameter of Suction and Delivery Pipes.	Height to which Water may be forced with 35 to 40 lbs. Boiler pressure	PRICE.
Inches.	Inches.	Inches.				Inches.	Inches.	Inches.	Feet.	£
4	2	5½	100	710	56	¾	1	1½	227	
6	3	8	70	1694	135	1	1½	3	227	
6	4	8	70	2710	216	1	1½	3	127	
5	5	8	60	3870	*	1	1½	4	60	
7	5	8	60	3870	309	1	1½	4	110	
7	5	10	60	4980	398	1	1½	4	110	
6	6	10	60	7200	*	1	1½	4	56	
8	6	10	50	5810	464	1½	2	4	100	
9	5	10	50	4120	330	1½	2	4	180	
7	5	15	50	5760	...	1	1½	4	110	
9	5	15	40	4800	...	1½	2	4	180	
8	6	15	50	8640	..	1½	2	4	100	
10	6	15	40	7200	...	1½	2	4	156	
8	8	20	50	20680	...	1½	2	6	56	
10	8	20	50	20680	...	1½	2	6	88	
12	7	20	40	12080	...	2	3	5	150	
14	7	20	40	12080	...	2½	3½	5	227	
15	8	20	40	16380	...	3	4	6	197	

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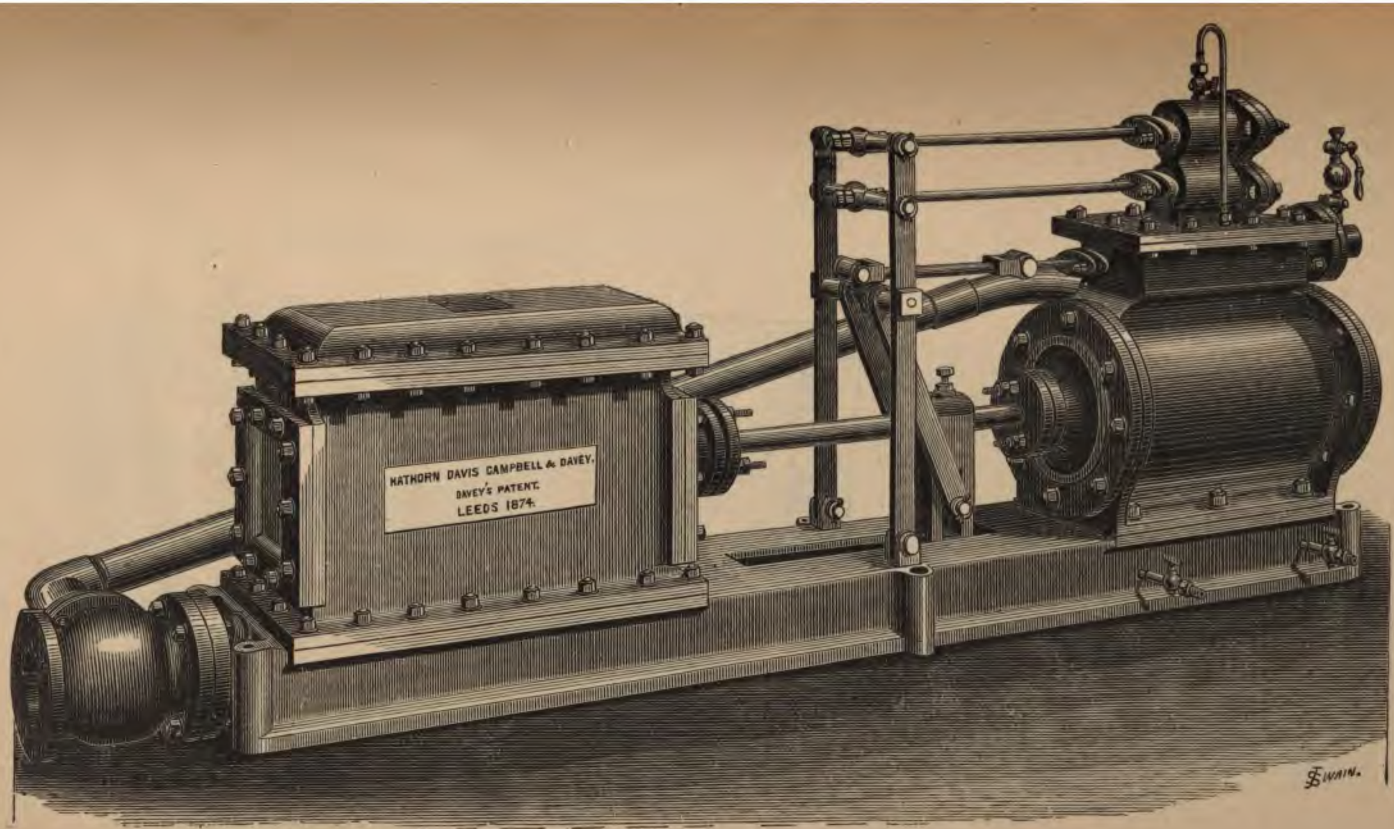
* A vacuum of from 8 to 10 lbs. can be obtained with the Condenser.



THE DIFFERENTIAL CONDENSING STEAM PUMP.
(DAVEY'S PATENT.)

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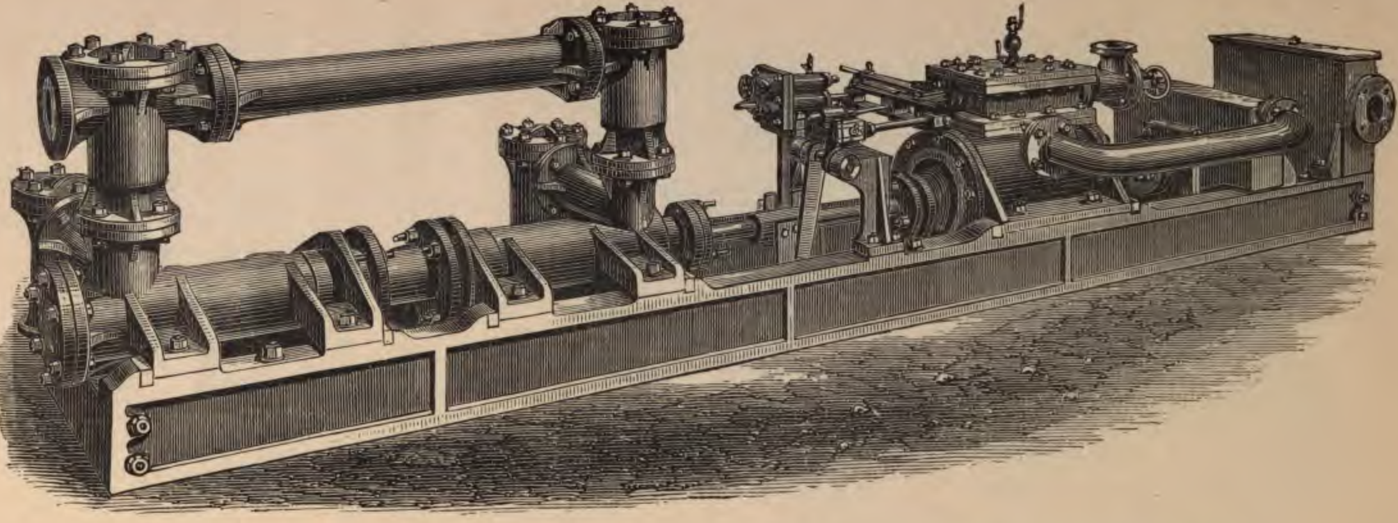
[illegible]EN F[illegible]ONS.



SINGLE CYLINDER CONDENSING DIFFERENTIAL PUMPING ENGINE AND PUMPS.
(**DAVEY'S PATENT.**)

SINGLE CYLINDER DIFFERENTIAL CONDENSING AND NON-CONDENSING PUMPING ENGINE AND DOUBLE RAM PUMPS.

Pumps of any size can be put to either of the undermentioned Engines.

Effective Horse-power, with 25 lbs. average effective pressure.	Diameter of Cylinder.	Length of Stroke.		Number of Strokes per Minute.	Diameter of Pump Ram	Number of Gallons per Minute.	Height of Lift.	Quantity of Injection Water required for Condenser.
	Inches.	Ft.	In.		Inches.		Feet.	
6	12	1	8	20	6	...	...	This varies with the power developed, 3½ gals. per effective horse-power per minute should be allowed.
7	12	2	0	20	6	90	250	
8	12	3	0	15	8	...	...	
10	14	2	0	20	6	...	...	
10	14	2	0	20	8	...	...	
11	14	3	0	15	9	...	...	
12	16	2	0	20	8½	...	...	
14	16	3	0	15	10	...	...	
17	*18	2	0	20	11	*300	135	
18	18	3	0	15	10	290	...	
20	18	4	0	13	...	...	...	
18	20	2	0	20	...	...	...	
22	†20	3	0	15	†4½	58	1000	
25	20	4	0	13	...	...	...	
23	22	2	0	20	8	160	400	
26	22	3	0	15	...	...	...	
30	22	4	0	13	...	...	...	
37	26	3	0	15	12½	450	240	
42	26	4	0	13	...	...	...	
50	28	4	0	13	...	...	...	
56	28	5	0	12	...	...	...	
49	30	3	0	15	9	230	480	
56	30	4	0	13	...	...	...	
88	35	5	0	12	...	...	...	
115	40	5	0	12	...	...	...	
193	‡52	5	0	12	‡9 & 11½	300 & 500	700 & 500	

* Surface Condenser to this Engine.

Non-Condensing Engine.

e 9-inch pump has a column of 700 feet, and the 11½ inch pump a column of 500 ft.

THE DIFFERENTIAL CORNISH ENGINE.

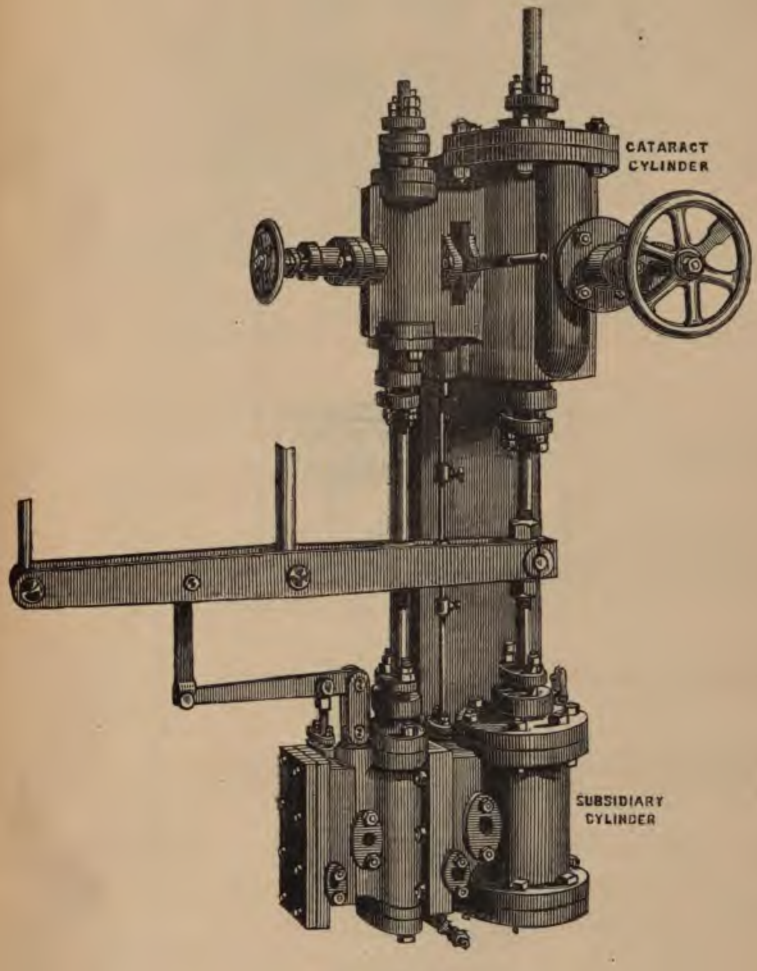
(Davey's Patent.)

From "Engineering."

The success which has attended the differential gear in the pumping engines specially designed by Mr. H. Davey, and known as differential engines, has led to its application to existing Cornish engines as a means of introducing into them an element of safety. A small variation of load on the veritable Cornish engine is attended with much danger, and the failure of the drawing lift or a spear rod frequently results in a greater disaster than the breaking of the spring beams. The main beam sometimes gives way under the shock, and cylinders are often broken. It is therefore a matter of great consequence to render the engine self-governing in order that these accidents may be averted. The accompanying engravings illustrate the gear as it has been made and applied by Messrs. Hathorn, Davis, Campbell, and Davey, of the Sun Foundry Company, Leeds, to several Cornish engines. The main principle of the gear is that the valves of the engine have a motion, the resultant of two other motions, the first an independent constant motion, and the other the motion of the engine, which may be termed a dependent variable motion. Any erratic movement of the engine resulting from a change of load, or other cause, produces a corresponding variation in the distribution of steam through the resultant motion of the valves.

The illustration on the opposite page represents a perspective view of the gear. It consists of a lever called the main lever, by means of which motion is given to the valves through a rod. The motion of the engine is given to the outer end of the lever, through the rod by means of a lever of the first order, the long end of which is attached to the plug rod, the other end deriving its motion from the subsidiary cylinder, and being controlled by means of the cataract. The cylinder has a slide valve which is worked by means of a tappet arm on the rod of the piston of a secondary cylinder; the motion of the secondary piston is also controlled by a secondary cataract. The slide valve is however free to move with the motion of the hand lever. It will be seen that there are two hand wheels and a hand lever attached to the cataracts. The function of the large wheel is to regulate the speed of the engine during the stroke; the small wheel is for regulating the pause between the strokes; whilst the hand lever enables the engineman to hand work the engines. The rocking shaft is employed to give motion to the valves of the engine in the usual way, clearly shown in the engraving.

The action of the gear may thus be described. Let the engine be "out doors." The engine end of the main lever will then be in its highest, and the opposite end in its lowest position, the secondary lever being lifted so as to admit steam to the bottom of the secondary cylinder. The engine will pause until the piston of the secondary cylinder shall have travelled to the end of its stroke, and have lifted the valve of the subsidary cylinder. The pause will be long or short, according to the regulation of the secondary cataract. The piston of the subsidiary cylinder then having steam on its lower surface will travel upwards, actuating the main lever with a speed *dependent on the adjustment* of the cataract. In doing so the steam valve of



THE DIFFERENTIAL CORNISH ENGINE GEAR.

(DAVEY'S PATENT.)

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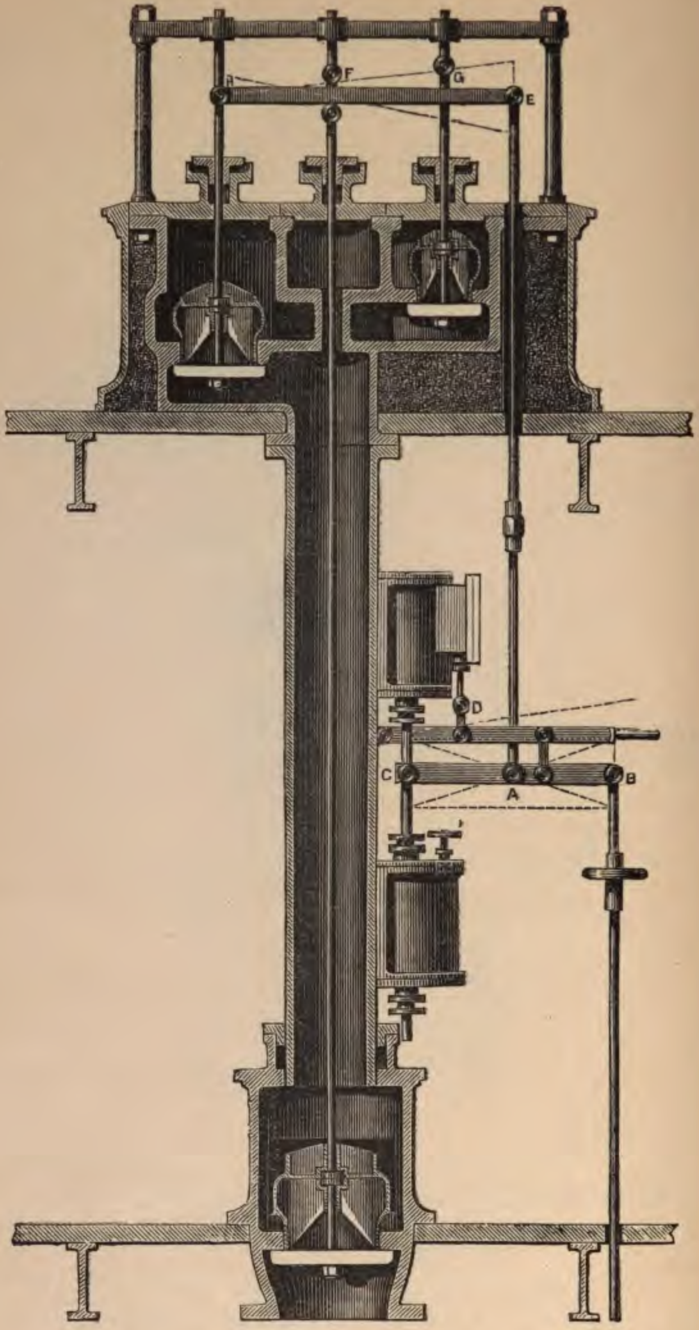


THE DIFFERENTIAL CORNISH ENGINE.

the engine will be opened through the medium of the rod, and the rocking shaft, &c., and will be opened quickly because the engine end of the lever is for the time being stationary. Steam now being admitted on the engine piston, it will, after overcoming the inertia of the load, move off at an increasing speed, which is communicated to the engine end of the main lever. The result is that as the opposite end of the lever is moving uniformly in the opposite direction at the same time, the motion of the centre is soon reversed in direction, and the steam valve which was opened by the motion of the subsidiary piston, is closed by the differential motion brought into action by the motion of the main piston acting on the same point.

It will be seen that the value of the differential will only be changed by a change in the motion of the engine, because the other component of the differential is a constant, and the time of closing the steam valve will be earlier or later as the differential is greater or less, and as changes in the motion of the engine are produced by variations of load and steam pressure, it therefore follows that these variations produce their correction in the distribution of steam.

If through loss of load the motion of the piston of the engine becomes excessively erratic, steam will, by the action of the gear, be admitted on the reverse side. The full load has often been suddenly thrown off from a differential engine moving at its normal speed without any damage resulting.

THE DIFFERENTIAL CORNISH ENGINE.

The illustration on page 38 represents the application of the Differential expansive gear to the ordinary Cornish engine, dispensing with the complicated gear usually applied, and at the same time rendering the engine safe in working under varying conditions of load and resistance. The *modus operandi* may thus be briefly described. The lever C.A.B. has a constant motion imparted to the end C by means of the subsidiary piston, governed by the cataract governor I, whilst the end B has a motion, taken from any convenient part of the engine, corresponding to the motion of the main steam piston on a reduced scale, and opposite in direction to the motion of the end C. If the motion of the end B is constant then will the motion of A be constant, but should the motion of B vary from a variation of load or pressure on the engine then will A* also vary, and as the distribution of steam is effected by the motion of A, the distribution will also vary with every minute erratic motion of the main steam piston. Following out the various motions produced on the steam eduction and equilibrium valves, it will be seen that every variation in the motion of the main steam piston produces its corresponding correction in the distribution of the steam. To take an example: should the steam pressure be increased on the piston, or the load of the engine decreased, then the point B will move at an increased velocity, whilst the motion of C remains at all times constant. The variation then affects the point A in such a manner, as to allow the steam valve to close earlier, which is the required correction; again, should the resistance to the falling load, during the equilibrium stroke, be diminished, then the load would tend to fall at an increased velocity, and in like manner varying the motion of A so as to partially close the equilibrium valve and prevent the load from exceeding its normal speed.

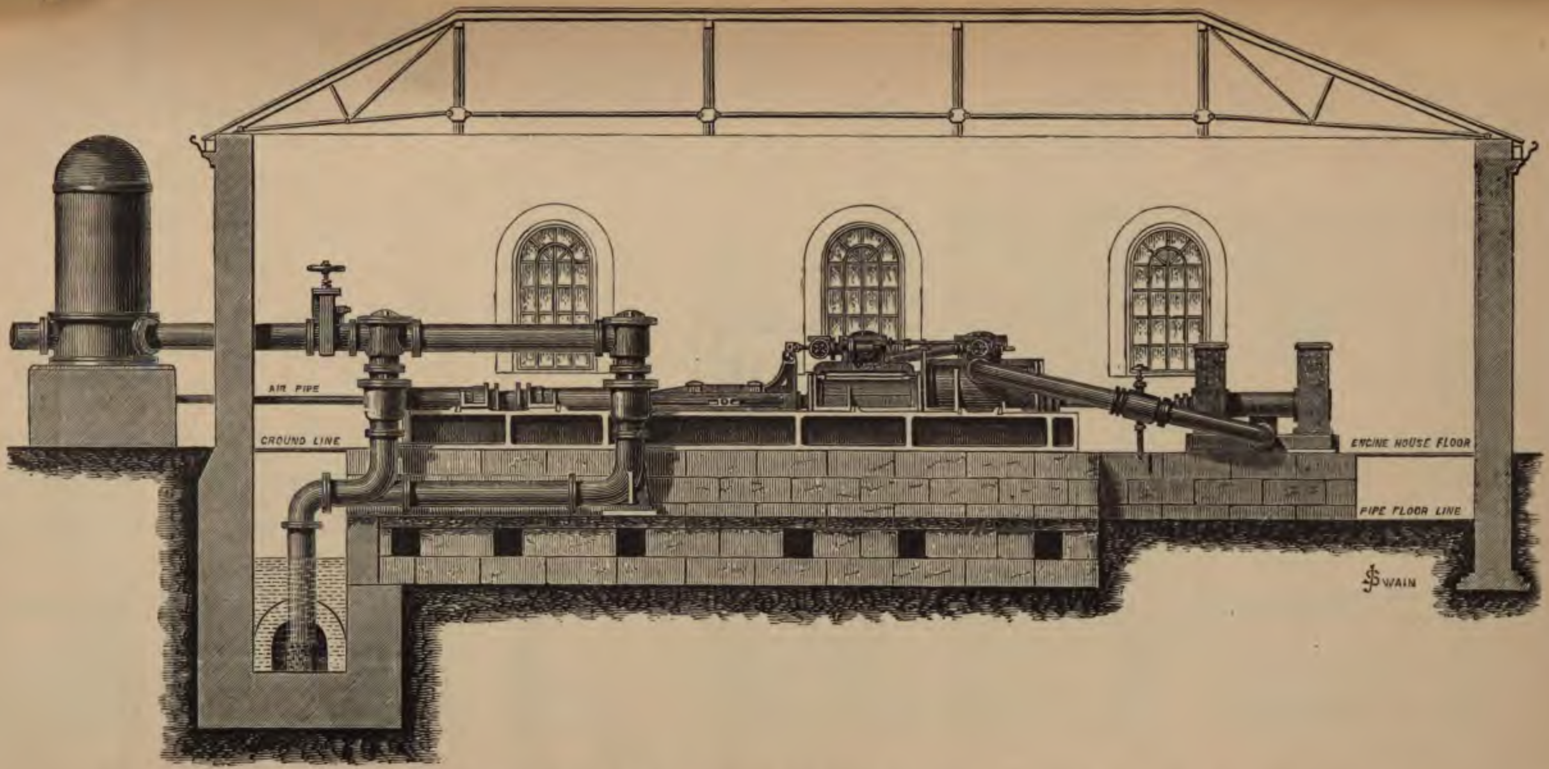
* Motion of A.

MEASURING WATER PUMPED FROM MINES, &c

It is very useful to be able to ascertain at any time the quantity of water which is being pumped. A convenient mode of doing it is to lead the water from the pump to a small pond in which the velocity of the stream may be rendered almost *nil*, and to allow the water from the pond to fall over a weir 18 inches wide with vertical sides. A vertical post should be driven in the pond, back from the weir or sill, in still water, and on the post should be painted a gauge in inches and fractions, with zero at the level of the sill. The following table will then show the quantity of water passing over the weir:—

WEIR TABLE FOR 18-INCH OVERFALL.

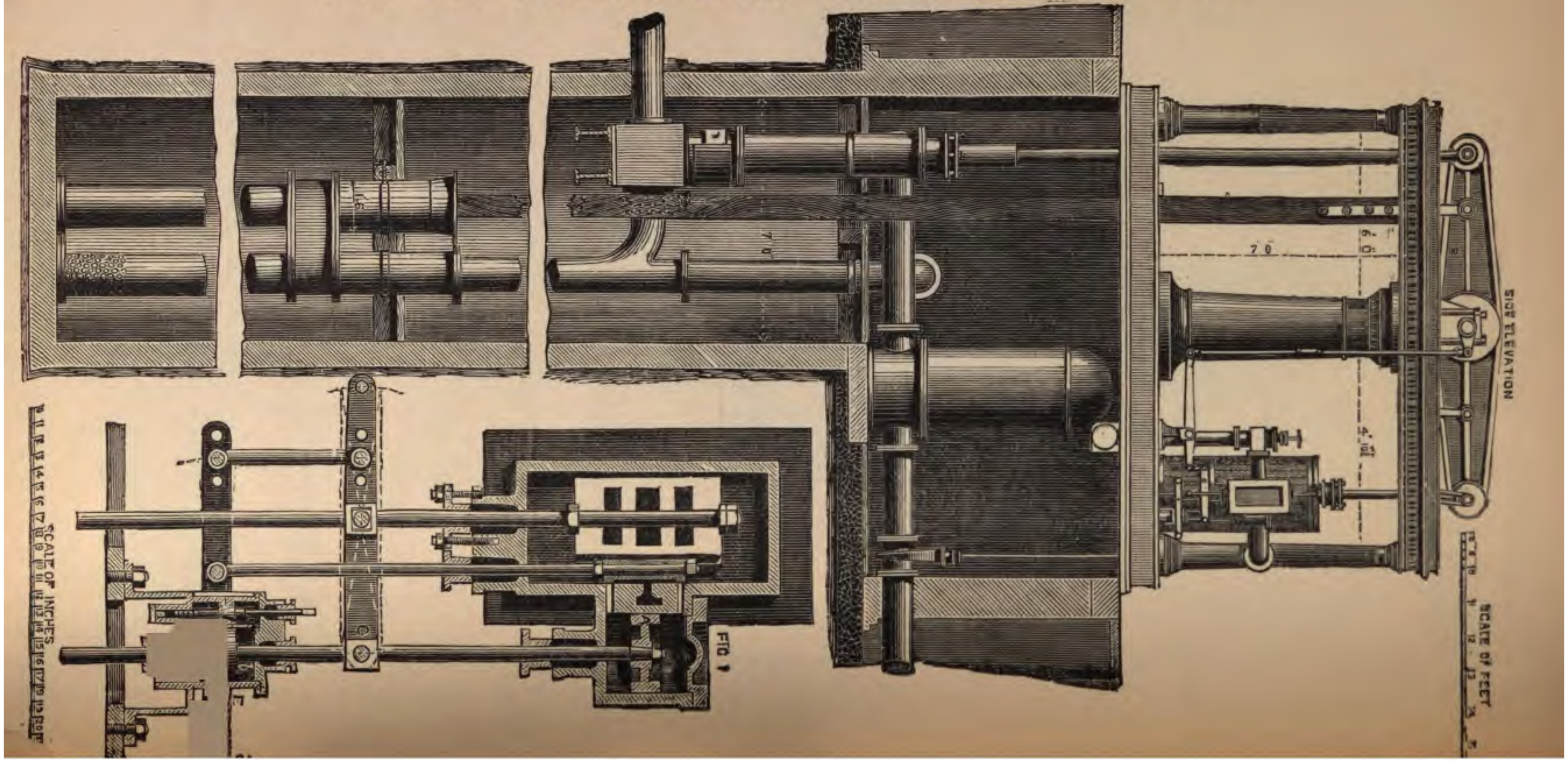
Depth of Water.	Cubic Feet per Minute.	Depth of Water.	Cubic Feet per Minute.	Depth of Water.	Cubic Feet per Minute.	Depth of Water.	Cubic Feet per Minute.
Inches.		Inches.		Inches.		Inches.	
1	7·725	7	142·974	13	361·296	19	639·216
1¼	10·730	7¼	150·637	13¼	372·583	19¼	652·445
1½	14·127	7½	158·55	13½	383·314	19½	665·122
1¾	17·833	7¾	167·04	13¾	393·866	19¾	678·030
2	21·770	8	174·627	14	404·296	20	690·168
2¼	26·016	8¼	183·	14¼	415·550	20¼	703·940
2½	29·490	8½	190·355	14½	426·495	20½	716·790
2¾	35·203	8¾	199·92	14¾	437·605	20¾	730·174
3	40·068	9	208·440	15	447·76	21	742·664
3¼	45·162	9¼	217·23	15¼	460·04	21¼	756·725
3½	50·488	9½	226·185	15½	471·375	21½	770·10
3¾	56·047	9¾	235·149	15¾	482·936	21¾	783·392
4	61·760	10	244·106	16	494·08	22	794·933
4¼	67·627	10¼	253·5	16¼	506·033	22¼	810·738
4½	73·648	10½	262·8	16½	517·729	22½	824·415
4¾	79·902	10¾	272·28	16¾	529·564	22¾	832·239
5	86·309	11	281·625	17	546·576	23	851·516
5¼	92·794	11¼	292·11	17¼	553·457	23¼	866·049
5½	99·610	11½	301·20	17½	565·470	23½	879·795
5¾	106·381	11¾	311·47	17¾	577·690	23¾	893·991
6	113·406	12	321·06	18	589·036	24	907·100
6¼	120·586	12¼	331·2	18¼	602·271	24¼	922·519
6½	127·920	12½	341·367	18½	614·601	24½	933·647
6¾	135·331	12¾	351·69	18¾	627·185	24¾	951·179

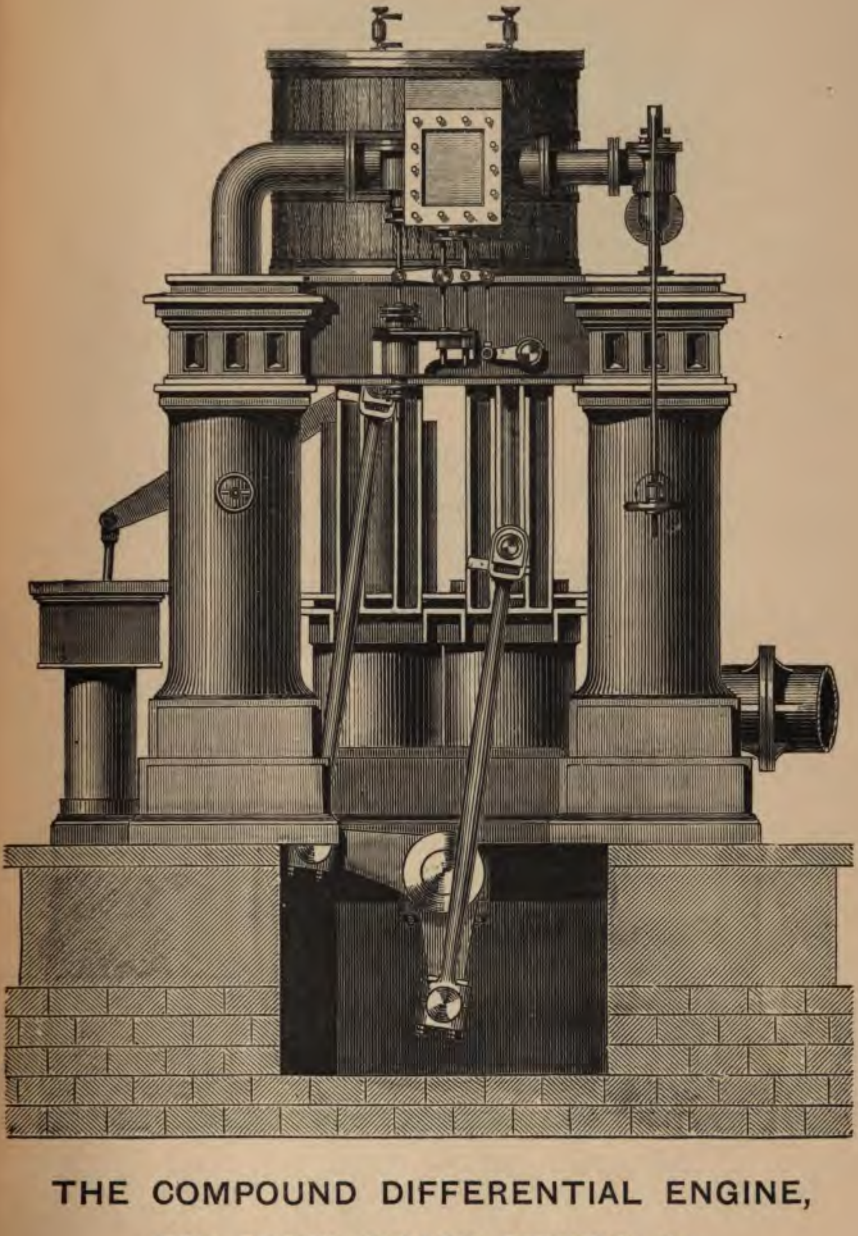


COMPOUND DIFFERENTIAL PUMPING ENGINE FOR TOWN WATER SUPPLY.

(DAVEY'S PATENT.)

THE DIFFERENTIAL BEAM ENGINE.—(DAVEY'S PATENT.)





THE COMPOUND DIFFERENTIAL ENGINE,

FOR TOWN WATER SUPPLY, &c.

(DAVEY'S PATENT.)

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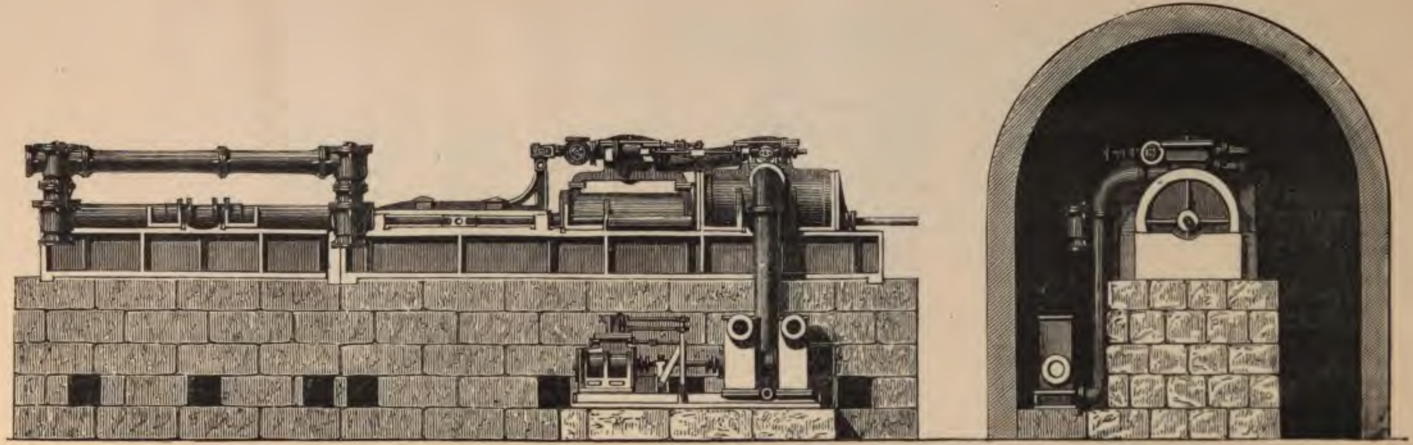

THE COMPOUND DIFFERENTIAL ENGINE AND FORCE PUMPS,

With Separate Condenser, as applied Underground.

(DAVEY'S PATENT.)

Horse-Power.

An actual horse-power equals 33,000 lbs. raised 1 foot high per minute, or 33,000 units of work, which I shall denote by the letters *H.-P.* The term nominal horse-power,—written *N. H.-P.*,—has a varied signification according to the fancies of different engineers, and is therefore made very confusing. It is a term which might be dispensed with, and the actual H.-P. universally adopted as the standard, to advantage.

The Admiralty rule for *N. H.-P.* is—

$$N.\ H.\text{-}P. = \frac{7AV}{33{,}000} \text{ or } \frac{D^2V}{600}$$

V = velocity of piston in feet per minute.
A = area of cylinder.
D = diameter of cylinder.
S = stroke of engine in feet.

Ordinary rules—

$$N.\ H.\text{-}P. = \frac{D^2\sqrt[3]{S}}{15{\cdot}6} \text{ for high-pressure engines.}$$

$$N.\ H.\text{-}P. = \frac{D^2\sqrt[3]{S}}{47} \text{ for condensing engines.}$$

For ordinary horizontal high-pressure non-condensing engines of commerce, an approximate rule is $N.\ H.\text{-}P. = \frac{D^2}{12}$, and $H.\text{-}P. = \frac{D^2}{6}$, which is approximately right for an initial pressure of 50 lbs. and cut off at half stroke, with 300 feet piston speed per minute.

Practical Notes and Formulæ for Pumps and Pumping Engines.

To find the quantity of water delivered from a given pump or pipe—

Let d = the diameter in inches, then :—

$\frac{d^2}{30}$ = the quantity delivered per foot, stroke, or flow in gallons, a little under the theoretical quantity.

A good working velocity of flow for water in the pipes is 200 feet per minute, and the speed of pump piston or ram most suitable for the "Differential Expansive" Pumping Engine, is equal to $\sqrt{L} \times 80$, where L = the length of the stroke in feet.

d = diameter of pipe in inches for 200 feet flow per minute.
p = diameter of plunger in inches—double-acting.
q = number of gallons delivered per minute.
$q = d^2 \times 6{\cdot}66 = 200\,\frac{d^2}{30}$, for a velocity of 200 feet per minute.

n = number of gallons delivered per hour.

$n = d^2 \times 400.$

$q = \frac{p^2}{30}\sqrt{L} \times 80.$

q = number of gallons of water raised per minute.

H = the height to which it is raised in feet.

$H.\text{-}P.$ = effective horse-power of engine without friction.

$H.\text{-}P. = qH \times 9\cdot000303 \quad = \quad \frac{qH}{3,300}$

h = head of water in feet.

p = pressure in lbs. per square inch.

f = pressure in pounds per square foot.

$p = h \times 0\cdot433.$

$h = p \times 2\cdot31.$

$f = h \times 62\cdot4.$

1 cubic foot of water = 6·24 gallons = 62·4 lbs. = ·557 cwt. = ·028 t
= $6\frac{1}{4}$ gallons approximately.

1 gallon = 10 lbs. or 0·16 cubic foot.

1 cwt. of water = 1·8 cubic foot = 11·2 gallons.

1 ton of water = 35·9 cubic feet = 224 gallons.

Weight of sea water = weight of fresh water × 1·028.

A cylinder of spring water 1 inch diameter and 1 fathom long
2·045 lbs.

Duty of Pumping Engines.

A cylinder of spring water, 1 in. diameter and 1 fathom long, weighs 2·045 lbs.

L = length of stroke.
N = number of strokes in one month.
H = height of lift in fathoms.
d = diameter of pumps.
q = number of bushels of coal consumed in one month.
Duty = number of lbs. lifted one foot high per bushel of coal.

$$Duty = \frac{[(H \times (d^2) \times 2{\cdot}045) \times N \times L]}{q}$$

The calculation was formerly made for bushels of coal, each weighing 94 lbs., but it is now usual to use the cwt. in the place of the bushel.

The usual mode of expressing the efficiency of a steam engine, is in terms of coal burnt per horse-power or units of work per hour, but as the efficiency of the boiler is not in such a case distinguished from that of the engine, it is important that a rule should be established by which the efficiency of either might be readily ascertained.

The efficiency of the boiler may be expressed in lbs. of water evaporated per lb. of fuel, whilst the efficiency of the engine should be denoted in *units of work done per* 1 *lb. of steam used.*

A formula for that purpose may be thus constructed :—

Let 28 = number of cubic inches of water in 1 lb. of steam.
I = initial pressure of steam.
S = its specific volume.
R = ratio of expansion employed.
a = average pressure per square inch $= \dfrac{(1 + Hyp.\ Log.\ R)\ I.}{R.}$
U = units of work done by 1 lb. of steam,

then $U = a \dfrac{(28 \times S \times R)}{12} =.$

Let I = 100 lbs. per square inch, then :
S = 270. See table, page 58.

The following table shows, in units of work, the comparative values of different degrees of expansion with a constant initial pressure equal to 100 lbs. per square inch; worked out by means of the above formula, which for 100 lbs. initial pressure stands thus : $U = a\ (630 \times R)$.

Initial Pressure.	Ratio of Expansion $= R$.	Units of Work.	Average Pressure $= \frac{(1 + \text{Hyp. Log. } R)\, I.}{R.}$
100 lbs.	0	63000·	100 lbs.
„ „	1·25	77017·50	97·8 „
„ „	1·66	94958·64	90·8 „
„ „	2·00	106596·	84·6 „
„ „	3·00	132111·	69·9 „
„ „	4·00	150192·	59·6 „
„ „	5·00	164115·	52·1 „
„ „	8·00	193536·	38·4 „
„ „	10·00	207900·	33·0 „

The formula for the duty of pumping engines* would then become $Duty = \frac{(H \times (d^2) \times 2{\cdot}045) \times N \times L}{W}$, in which W = weight of water in lbs., and *duty* = units of work per lb. of steam.

For the convenience of readily and accurately recording the duty of pumping engines, the water should be measured into the boiler by a meter made to register in lbs., and if at the same time the weight of fuel burnt were noted, the *duty* of both engine and boiler could be ascertained. The *duty* of the boiler would be found by dividing the number of lbs. of water evaporated by the number of lbs. of standard fuel burnt, and the results obtained should then be expressed thus:

Duty of engine = number of units of work per lb. of water evaporated.
Duty of boiler = number of lbs. of water evaporated per lb. of coal burnt.

Water Works.

CONSUMPTION OF WATER IN TOWNS.

15 to 20 gallons per head of population per day in non-manufacturing towns.

20 to 30 gallons per head in manufacturing towns.

Maximum demand $2\frac{1}{2}$ times the average.

Impounding reservoirs should contain 120 days' supply in the rainy districts and 200 days' supply in the less rainy districts of England.

Service reservoirs should contain 3 days' supply.

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* See page 51.

RAIN-FALL.

Greatest rain-fall in England in 24 hours about 3 inches.
Annual rain-fall in England from 20 to 70 inches.
Mean „ „ „ 42 inches.
Mean daily evaporation in England, ·08.

Infiltration in England in Winter, 33 per cent.
, „ Spring, 35 „
„ „ Summer, 52 „
„ „ Autumn, 48 „
Average for the year, 42 „

WELLS IN VARIOUS GEOLOGICAL FORMATIONS.

In the New Red Sandstone.—Robert Stephenson found when experimenting with wells in this formation around Liverpool, that abundance of water is stored up in the new red sandstone, but the sandstone is generally very pervious, admitting of deep wells drawing their supply from distances exceeding one mile; the permeability is however occasionally interfered with by faults or fissures filled with argillaceous matter. He also reported that there was no possibility of obtaining permanently more than about 1,000,000 or 1,200,000 gallons per day from any one well, and this only when not interfered with by other wells.

Through Chalk into Lower Green Sand.—A remarkable artesian well has been bored at Passy in this formation, for supplying water to Paris. The first water-bearing strata was reached at a depth of 1,894 feet, but the water did not rise to the surface. At length a true artesian spring was tapped at a depth of 1,923 feet, yielding 5,582,000 gallons per day.

Through Tertiaries into Upper Green Sand and Chalk.—Wells have been bored into this formation with varying success. At Brighton, a boring is said to yield 1,000,000 gallons per day. At Worthing, 30 feet above the sea and 360 feet deep in the chalk, a well gives a good supply. In the valley of the Lee wells have been successful up to 100,000 gallons per day near the surface. At Southampton, a depth of 1,317 feet was reached without success and the scheme was abandoned; several other similar undertakings met with a like result.

Domestic Wells generally catch the adjacent percolating water from the surface, and are very liable to receive impurities, unless great care is exercised in selecting the site and insuring a proper construction.

DIMENSIONS AND WEIGHT OF PUMP PIPES FOR MINES.

LENGTH OF PIPE, 9 FT.

Diameter of Pipe in inches.	Thickness in inches.	Number of Holes in Flange.	Size of Holes.	Distance between Holes.	WEIGHT.			Diameter of Pipe in inches.	Thickness in inches.	Number of Holes in Flange.	Size of Holes.	Distance between Holes.	WEIGHT.		
					Cwt.	Qrs.	Lbs.						Cwt.	Qrs.	Lbs.
4	¾	4	1⅛	8½	3	0	0	15	1¼	8	1½	20	18	3	0
4½	¾	4	1⅛	9	3	1	14	16	1¼	8	1½	21	19	0	0
5	⅞	6	1⅛	9½	4	1	14	16	1⅜	8	1½	21	22	0	0
6	1	6	1¼	10½	6	0	0	17	1¼	8	1½	22	21	1	0
7	1	6	1¼	11½	7	0	0	17	1⅜	8	1½	22	23	1	14
8	1	6	1¼	12½	8	0	0	18	1¼	10	1½	23	22	2	0
9	1	6	1⅜	14	9	0	0	18	1⅜	10	1½	23	24	3	0
9	1⅛	6	1⅜	14	10	0	0	19	1¼	10	1½	24	23	3	0
10	1	6	1⅜	15	10	0	0	19	1⅜	10	1½	24	26	0	14
10	1⅛	6	1⅜	15	11	1	0	20	1¼	10	1½	25	25	0	0
11	1	6	1⅜	16	11	0	0	20	1½	10	1½	25	30	0	0
11	1⅛	6	1⅜	16	12	1	14	21	1¼	10	1½	26	26	1	0
12	1⅛	6	1⅜	17	13	2	0	21	1½	10	1½	26	31	2	0
12	1¼	6	1⅜	17	15	0	0	22	1¼	10	1½	27	27	2	0
13	1⅛	8	1⅜	18	14	2	14	22	1½	10	1½	27	33	0	0
13	1¼	8	1⅜	18	16	1	0	23	1¼	12	1½	28	28	0	0
14	1⅛	8	1⅜	19	15	3	0	23	1½	12	1½	28	34	2	0
14	1¼	8	1⅜	19	17	2	0	24	1¼	12	1½	29	30	0	0
15	1⅛	8	1½	20	16	3	14	24	1½	12	1½	29	36	0	0

DISCHARGE OF WATER THROUGH PIPES.

180 Feet per Minute.			200 Feet per Minute.			220 Feet per Minute.		
Diameter of Pipe.	Head required per 100 Feet.	Gallons per Minute.	Diameter of Pipe.	Head required per 100 Feet.	Gallons per Minute.	Diameter of Pipe.	Head required per 100 Feet.	Gallons per Minute.
1	4·32	6·110	1	5·33	6·796	1	6·45	7·482
2	2·16	24·441	2	2·67	27·184	2	3·22	29·928
3	1·44	55·117	3	1·78	61·227	3	2·15	67·338
4	1·08	97·888	4	1·33	108·738	4	1·61	119·587
5	0·86	152·944	5	1·07	169·966	5	1·29	186·987
6	0·72	220·282	6	0·89	244·786	6	1·08	269·289
7	0·62	299·903	7	0·76	333·198	7	0·92	366·493
8	0·54	391·682	8	0·67	435·203	8	0·81	478·723
9	0·48	495·807	9	0·59	550·924	9	0·72	606·042
10	0·43	612·089	10	0·53	680·113	10	0·64	748·137
11	0·39	740·718	11	0·49	823·020	11	0·59	905·322
12	0·36	881·629	12	0·45	979·518	12	0·54	1077·408
13	0·33	1034·386	13	0·41	1149·110	13	0·50	1264·458
14	0·31	1199·614	14	0·38	1333·043	14	0·46	1466·472
15	0·29	1377·311	15	0·36	1530·069	15	0·43	1683·450
16	0·27	1566·855	16	0·33	1740·812	16	0·40	1915·392
17	0·26	1769·493	17	0·31	1965·895	17	0·38	2162·991
18	0·24	1983·333	18	0·30	2203·449	18	0·36	2424·168
20	0·22	2448·484	20	0·27	2720·320	20	0·32	2992·176
22	0·20	2962·872	22	0·24	3292·080	22	0·29	3620·664
24	0·18	3525·892	24	0·22	3917·450	24	0·27	4309·008

WATER WHEELS.

Let it be required to find the effective thrust of the crank of an overshot water wheel employed for pumping.

c = length of crank.

r = radius of wheel.

$\frac{r - c}{3}$ = effective leverage.

n = number of buckets.

W = weight of water in each bucket.

$$f = \text{friction} = \frac{\left(\frac{n}{3} \times W\right) \times \frac{r - c}{3}}{5}$$

p = effective thrust.

$$p = \left[\left(\frac{n}{3} \times W\right) \times \frac{r - c}{3}\right] - f.$$

DISCHARGE OF WATER OVER SILLS.

d = depth from surface of water where at rest, to the top of sill in inches.
c = cubic feet of water discharged per minute for every foot in width of sill.
$c = 5{\cdot}15\ \sqrt{d^3}$,
$H\ \ \frac{d}{12}$ = depth in feet.
$c = 214\ \sqrt{H^3 + {\cdot}035\ V^2\ H}$, if the water approaches the sill with a velocity = V.

PIPE JOINTING.

The weight of lead required for jointing socket pipes may be approximately estimated by reckoning 1 lb. per inch diameter of pipe for each joint up to 8 inches diameter; 1¼ lbs. from 8 inches to 16 inches diameter, and 1½ lbs. from 16 inches to 24 inches diameter. In the small pipe joints the thickness of lead should be about $\frac{5}{16}$ inch, and in the large ones about $\frac{7}{16}$ inch. The depth of socket is usually about 4½ inches, the pipe measuring 9 feet 4½ inches over all.

Strength of cylinders—circumferential strain:

r = internal radius.
p = pressure per square inch.
c = cohesive strength of the metal.
x = the required thickness.

$$x = \frac{p \times r}{c - p}.$$

PROPORTIONS OF CAST-IRON FLANGE PIPES.

Diameter of Pipe.	Diameter of Flange.	Thickness of Flange.	Number of Bolts.	Diameter of Bolts.	Diameter of Circle of Bolts.
1½ inches	4½ inches	½ inch	3	⅜ inch	3¼ inches
2 ,,	5¼ ,,	½ ,,	3	$\frac{7}{16}$,,	3¾ ,,
2½ ,,	6 ,,	⅝ ,,	4	$\frac{7}{16}$,,	4½ ,,
3 ,,	6½ ,,	⅝ ,,	4	½ ,,	5 ,,
4 ,,	8 ,,	⅝ ,,	4	$\frac{9}{16}$,,	6¼ ,,
5 ,,	9¼ ,,	¾ ,,	4	$\frac{9}{16}$,,	7½ ,,
6 ,,	10½ ,,	¾ ,,	6	⅝ ,,	8¾ ,,
7 ,,	12 ,,	¾ ,,	6	⅝ ,,	10 ,,
8 ,,	13¼ ,,	⅞ ,,	6	⅝ ,,	11¼ ,,
9 ,,	14½ ,,	⅞ ,,	6	⅝ ,,	12¼ ,,
10 ,,	16 ,,	1 ,,	6	¾ ,,	13½ ,,
12 ,,	18¼ ,,	1 ,,	6	¾ ,,	16 ,,

WEIGHTS AND PROPORTION OF CAST-IRON SOCKET PIPES, FOR A HEAD OF ABOUT 200 FEET.

Diameter of Pipe.	Length without Socket.		Depth of Socket.	Lead Joint.			Average Weight of the Pipe.			Average Weight of Quarter Bends.		
Inches.	Ft.	Ins.	Inches.	Thickness.	Depth.	Weight.	Cwt.	qrs.	lbs.	Cwt.	qrs.	lbs.
1½	6	0	3	¼ inch	1½ inch	1·2 lbs.	0	1	14	0	1	0
2	6	0	3	¼ "	1½ "	1·4 "	0	2	0	0	1	2
2½	6	0	3¼	¼ "	1½ "	1·6 "	0	2	11	0	1	9
3	9	0	3½	¼ "	1¾ "	2·3 "	1	0	8	0	1	17
4	9	0	4	$\frac{5}{16}$ "	2 "	4·0 "	1	2	0	0	2	19
5	9	0	4	$\frac{5}{16}$ "	2 "	5·0 "	2	0	0	0	3	0
6	9	0	4¼	$\frac{5}{16}$ "	2¼ "	6·5 "	2	2	0	0	3	3
7	9	0	4¼	$\frac{5}{16}$ "	2¼ "	7·7 "	3	0	12	1	0	25
8	9	0	4¼	$\frac{5}{16}$ "	2¼ "	8·2 "	3	2	24	1	2	2
9	9	0	4½	$\frac{5}{16}$ "	2½ "	10·4 "	4	1	0	1	2	18
10	9	0	4½	$\frac{5}{16}$ "	2½ "	11·5 "	4	3	14	3	3	0
12	9	0	4½	$\frac{5}{16}$ "	2¾ "	15·0 "	6	2	0	3	3	21

Weight of Pipes.

D = outside diameter in inches.
d = inside diameter.
W = weight of lineal foot in lbs.
$W = k\,(D^2 - d^2)$.

$k = 2{\cdot}45$ for cast-iron.
$= 2{\cdot}64$ for wrought-iron.
$= 2{\cdot}82$ for brass.
$= 3{\cdot}03$ for copper.
$= 3{\cdot}86$ for lead.

HYDRAULIC FORMULÆ.

g = force of gravity = 32·2.
H = head of water in feet.
P = pressure per square inch.
v = theoretical velocity in feet per second.
$H = P \times 2{\cdot}307.$ $\quad P = H \times {\cdot}4335.$
$\sqrt{2g} = 8{\cdot}025.$ $\quad V = 8\ 025\sqrt{H}.$
$H = {\cdot}0155\,v^2.$

DELIVERY OF WATER IN PIPES.

D = diameter of pipe in inches.
H = head of water in feet.
L = length of pipe in feet.
W = cubic feet of water discharged per minute.

$$W = 4{\cdot}72\sqrt{\frac{D^5 H}{L}}$$

$$D = 538\sqrt[5]{\frac{L W^2}{H}}$$

SATURATED STEAM.			HYPERBOLIC LOGARITHMS.		
Absolute pressure in lbs. per square inch.	**Temperature of Steam—Fah.**	**Specific Volume.**	**Ratio of Expansion = R.**	**Hyp. Log.**	**Portion of stroke at which steam is cut off.**
5	162·3	4527	1·25	·223	$\frac{8}{10}$
10	193·3	2358	1·43	·358	$\frac{7}{10}$
*14·7	212·0	1642	1·66	·506	$\frac{6}{10}$
20	228·0	1229	2·00	·693	$\frac{1}{2}$
23	235·5	1075	2·50	·916	$\frac{4}{10}$
30	250·4	838	3·33	1·202	$\frac{3}{10}$
35	259·3	726	4·00	1·386	$\frac{1}{4}$
40	267·3	640	5·00	1·609	$\frac{1}{5}$
45	274·4	572	6·00	1·791	$\frac{1}{6}$
50	281·0	518	7·00	1·945	$\frac{1}{7}$
55	287·1	474	8·00	2·079	$\frac{1}{8}$
60	292·7	437	9·00	2·197	$\frac{1}{9}$
65	298·0	405	10·00	2·302	$\frac{1}{10}$
70	302·9	378			
75	307·5	353			
80	312·0	333			
85	316·1	314			
90	320·3	298			
95	324·1	283			
100	327·9	270			

The above table of Hyperbolic Logarithms is given for the convenience of calculating the average pressure for a given initial pressure and ratio of expansion.

R = ratio of expansion.
I = initial pressure.
a = average pressure throughout the stroke.

$$a = \frac{(1 + Hyp.\ Log.\ R)\ I}{R}$$

Condensing Water.

W = weight of steam to be condensed.
T = its temperature.
l = its latent heat.
W' = weight of injection water required.
t = its temperature.
t' = temperature of the mixture.

$$W' = \frac{(T + l - t')\,W}{t' - t}.$$

For values of l, see Table on page 59.

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* Atmospheric pressure.

Pressure of Steam in lbs.	Latent Heat of Steam.	Sum of Latent and Sensible Heat.	Tempera-ture—FAH.	Pressure of Steam in lbs.	Latent Heat of Steam.	Sum of Latent and Sensible Heat.	Tempera-ture—FAH.
	1092·6	1124·6	32°	29	939·6	1187·6	248°
	1080·0	1130·0	50°	39	927·0	1193·0	266°
	1067·4	1135·4	68°	53	914·4	1198·4	284°
	1054·8	1140 8	86°	69	901·8	1203·8	302°
1	1042·2	1146·2	104°	90	889·2	1209·2	320°
2	1029·6	1151·6	120°	115	874·8	1212·8	338°
3	1017·0	1157·0	140°	146	862·2	1218·2	356°
4·5	1004·4	1162·4	158°	182	849·6	1223·6	374°
7	991·8	1167·8	176°	229	835·2	1227·2	392°
10	979·2	1173·2	194°	279	822·6	1232·6	410°
[*]14·706	966·6	1178·6	[†]212°	348	808·2	1236·2	428
21	952·2	1182·2	230°	402	795·6	1241·6	446

WEIGHT OF BOILERS.

Reckon 6 lbs. per ⅛″ of thickness of plate for each superficial foot. This will give the weight very near, including lap and rivets.

STRENGTH OF BOILERS.

Strength of plate = 100.
Strength of single rivetted joints =56.
Strength of double rivetted joints = 70.

STRENGTH OF BOILER TUBES.

P = collapsing pressure in lbs.
K = thickness of plate in inches.
L = length of tube in feet.
D = diameter of tube in inches.

$$P = 806300 \frac{K \times 2{\cdot}19}{L \times D}; \text{ or}$$

$$Log.\ P = 1{\cdot}5265 + 2{\cdot}19\ log.\ 100\ K - log.\ (L\ D).$$

CHIMNEYS.

F = number of lbs. of coal consumed per hour.
h = height of chimney in feet.
A = area of chimney at top in square inches.

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[*] Atmospheric pressure.

[†] Boiling point.

$H.\text{-}P.$ = indicated horse-power.

$$A = \frac{15\,F}{\sqrt{h}};$$

for economical pumping engines—

$$A = \frac{70\,H.\text{-}P.}{\sqrt{h}}.$$

H.P.	Height of Chimney.	Diameter at Base.	Diameter of Flue.
50	75 feet.	8 feet.	2 feet 4 inches.
100	100 „	11 „	3 „ 0 „
200	120 „	13 „	4 „ 0 „
300	160 „	14 „	4 „ 6 „

VELOCITY OF ARTIFICIAL DRAUGHT.

H = height of chimney in feet.
T = temperature of air supplying the chimney.
t = temperature of air at top of chimney.
V = velocity in feet per second.

$$V = 36{\cdot}5\,\sqrt{H\,(T-t)}.$$

SHORT LINK CHAINS—BEST TESTED.

Size.	Working Load.	Outside Dimensions.
$\frac{3}{8}$ inch.	1 ton.	$1\frac{7}{8}$ inches × $1\frac{1}{4}$ inches.
$\frac{7}{16}$ „	$1\frac{1}{2}$ „	$2\frac{1}{16}$ „ × $1\frac{7}{16}$ „
$\frac{1}{2}$ „	2 „	$2\frac{1}{4}$ „ × $1\frac{3}{4}$ „
$\frac{9}{16}$ „	3 „	$2\frac{7}{16}$ „ × $1\frac{7}{8}$ „
$\frac{5}{8}$ „	4 „	$2\frac{3}{4}$ „ × $2\frac{1}{8}$ „
$\frac{11}{16}$ „	5 „	$3\frac{1}{16}$ „ × $2\frac{3}{8}$ „
$\frac{3}{4}$ „	6 „	$3\frac{3}{8}$ „ × $2\frac{1}{2}$ „
$\frac{13}{16}$ „	7 „	$3\frac{3}{4}$ „ × $2\frac{13}{16}$ „
$\frac{7}{8}$ „	8 „	$4\frac{1}{16}$ „ × 3 „
$\frac{15}{16}$ „	9 „	$4\frac{3}{8}$ „ × $3\frac{3}{16}$ „
1 „	10 „	$4\frac{5}{8}$ „ × $3\frac{1}{2}$ „
$1\frac{1}{16}$ „	$10\frac{1}{2}$ „	$4\frac{7}{8}$ „ × $3\frac{3}{4}$ „
$1\frac{1}{8}$ „	12 „	$5\frac{3}{16}$ „ × $4\frac{1}{16}$ „

Table of Observations upon Ten Cornish Engines. By John S. Enys, Esq., 1838.

		CONSOLIDATED MINES.						UNITED MINES.			
		Taylor's Engine.	Davey's Engine.[a]	Job's Engine.[b]	Woolf's Engine.[c]	Bawden's Engine.	Pearce's Engine.[d]	Cardozo's Engine.[e]	Eldon's Engine.[f]	Loam's Engine.[g]	Hocking's Engine.
DIMENSIONS OF ENGINE.	Diameter of Cylinder ... in inches ...	85	80	65	90	90	65	90	30	85	85
	Stroke of Piston ... ... in feet ...	10	11⅓	9	10	10	9	9	9	10	10
	Diameter of Steam Valve in inches ...	12	13	9	8	8	7	10	5	10	12
	Ditto of Equilibrium Valve ,, ...	16	18	12	16	16	12	13	7	16	16
	Ditto of Exhaust Valve ... ,, ...	20	24	14	19	19	14	15	10	19	19
DIMENSIONS OF BOILERS.	Number of Boilers	4	3	2	4	3	3	3	1	3	3
	Length of Boilers ... ... in feet ...	3.36 1.40	37	31 32	35	36	36	36	36	1.32 2.38	44
	Diameter of Boilers ... ... ,, ...	6½	7	6½	6¼	6¼	6¼	6¼	6½	6½	6½
	Ditto of Tubes ... ... ,, ...	3. 3¾ 1. 4	4⅓	3¾	3¾	3¾	3¾	3¾	4	4	4
	Length of Fire Bars ... ... ,, ...	4	4	4	4	4	4	4	4	4	4
	Total Area of Fire Bars ... ,, ...	63	52	30	60	45	45	48	16	18	48
	Heating Surface Exposed to Flues in feet	3781	3151	1598	3481	2694	2694	2694	941	2952	3451
	Water Space ... ... in cubic feet	2467	2025	1033	2140	1650	1650	1650	579	1706	2085
	Steam Space ... ... ,,	735	580	315	608	468	468	468	178	528	645
TEMPERATURES OBSERVED.	Open Air ... ... Fah.	57°	57°	—	57°	57°	51°	55°	56°	51°	55°
	Engine House ... ... ,, ...	—	—	—	—	—	67°	—	65°	63°	66°
	Ashes over Boiler ... ... ,, ...	80°	80°	81°	98°	98°	97°	88°	99°	79°	82°
	Cylinder Cover ... ... ,, ...	111°	90°	102°	126°	96°	109°	94°	98°	90°	84°
	Middle of Cylinder Clothing* Fah. ...	W 77°	W 76°	B 95°	B 102°	B 140°	—	B 79°	W 60°	W 67°	W 68°
	Clothing of Steam Pipe ... Fah.	79°	—	—	130°	95°	—	140°	97°	80°	82°
	Condensing Water ... .. ,, ...	64°	84°	58°	115°	110°	57°	60°	61°	63°	60°
	Hot Well ... ... ,, ...	98°	100°	91°	140°	140°	97°	104°	94°	102°	96°
Height of Condenser Barometer... ... ... in inches ...		27¼	27½	—	25½	—	—	—	—	—	27½
Number of Plunger Pumps		9	12	2	7	8	9	8	1	5	5
Number of Bucket Pumps		2	2	—	1	2	2	2	—	4	3
Water Load per square inch of Piston ... ... in lbs. ...		11.46	13.12	8.78	11.56	8.3	16.8	11.5	17.96	11.95	13.58
Proportion of Stroke where Steam cut off		⅕	⅙	—	⅛	⅛	¼	⅛	⅛	2/11	—
Strokes per minute		8½	7½	—	9	9	10	8½	9	8	7
Proportion of duration of in-door to out-door Stroke		4 : 7	5 : 8	4 : 7	5 : 7	5 : 7	5 : 5	4 : 6½	4 : 7	1 : 2	4 : 7½
Grease used per day ... ... in lbs. ...		12	12	10	12	12	10	12	6	12	12
Oil used per day ... ... in pints ...		1	1	1	1	1	1	1	½	1	1
Men Employed		4	4	3	4	4	3	4	3	4	4
Boys Employed		3	3	3	4	4	3	3	—	3	3

(a) Six years old. The Show Engine of the Mine. (b) Employed in raising water forty-seven fathoms for the wheels and injection. (c) Cylinder in bad condition; since changed. (d) Old Engine; 1820. (e) Old Engine; no steam jacket. (f) Old Engine refitted. (g) An old 90-inch Cylinder used for the steam jacket of this engine.

* The prefix of the letter B signifies that the casing is of Brick; and W of Wood.

WEIGHT OF WROUGHT-IRON.

The weight of 1 cubic inch of wrought-iron = 0·28 lbs.

$$\frac{\text{Number of cubic inches}}{8000} = \text{tons.}$$

$$\frac{\text{Number of cubic inches}}{400} = \text{cwts.}$$

$$\frac{\text{Number of cubic inches}}{100} = \text{qrs.}$$

$$\frac{\text{Number of cubic inches}}{3{\cdot}5} = \text{lbs.}$$

WEIGHT OF CAST-IRON.

The weight of 1 cubic inch of cast-iron = 0·26 lbs.

$$\frac{\text{Number of cubic inches}}{8640} = \text{tons.}$$

$$\frac{\text{Number of cubic inches}}{432} = \text{cwts.}$$

$$\frac{\text{Number of cubic inches}}{108} = \text{qrs.}$$

$$\frac{\text{Number of cubic inches}}{3{\cdot}85} = \text{lbs.}$$

1 cubic foot of cast-iron = 448 lbs. = $\frac{1}{5}$th ton = 4 cwts.

1 foot superficial 1 inch thick = $\frac{1 \text{ cwt.}}{3}$

Let n = number of cubic inches of cast-iron,
W = weight in lbs.

$W = \frac{n}{4}$ + (4 × the number of hundreds expressed by the first, or hand figure in product), thus: Let $n = 1728$ cubic inches, then $= \frac{1}{}$ $= 432$ and $4 \times 4 = 16 \therefore W = 432 + 16 = 448$ lbs.

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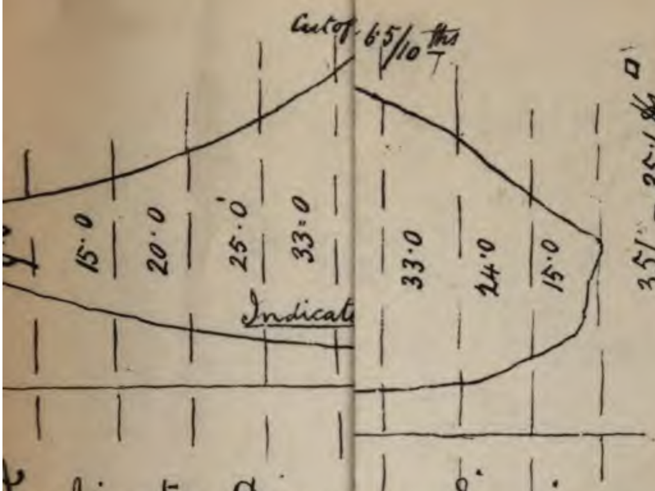

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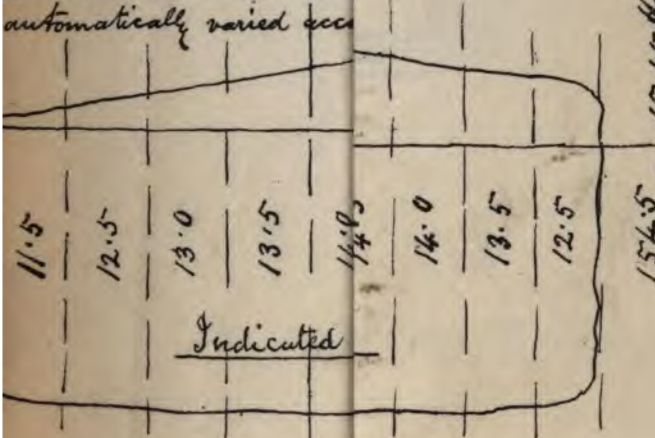

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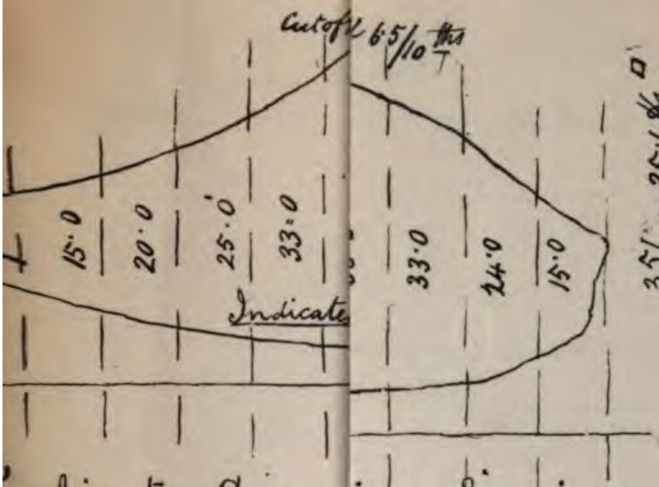

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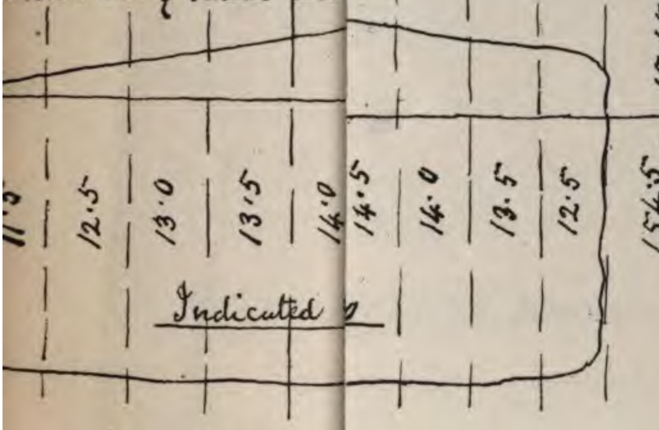

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